AMERICAN ENGINEER AND RAILROAD JOURNAL.

JUNE, 1899.

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STANDARD CONSOLIDATION FREIGHT LOCOMOTIVES.

PENNSYLVANIA RAILROAD.

Classes H5 and H6.

WITH AN INSET.

No locomotives ever built in this country or abroad have been designed with more care and intelligence or with the benefit of wider experience than the heavy freight engines, Classes H5 and H6, of the Pennsylvania Railroad.

These two consolidation designs are for entirely different purposes. H5 is a special pushing engine for mountain service at Altoona, where maximum power is required for a short distance only. Fifteen of these are in service, and, as far as possible, the same details were used in H6, which is a road engine, and was not required to be as powerful. The differences are seen in the drawings and dimensions, Figs. 2 and 3 of the inset accompanying this issue. The boilers are the same except at the back ends, and the details shown in the drawings apply generally to both. The differences in appearance between the two designs are not marked, even when they are seen standing together. The general plan of each design is shown and the details presented in the drawings are nearly all of H6, and are practically the same for both.

The locomotives of both classes now in service have been built at the Juniata shops and an order for 25 of the H6 is now in hand at the Baldwin Locomotive Works. The preliminary work was so thoroughly and well done that road experience with both designs has not developed a defect or shown a single desirable change to be embodied in the ones now building. There are few locomotives built under present conditions of which this may be said. Fig. 1 is from a photograph of engine No. 1,431, one of the H5 class, which shows the general appearance of both. For such large engines they are remarkably handsome.

Of Class H6 we give the following general dimensions:

Cylinders	23½x28 in.
Cylinders, spread	90 in.
Tubes, number	306
Tubes, outside diameter	2 in.
Tubes, length between sheets	14 ft.
Firebox, inside	10 ft. x 40 in.
Heating surface tubes	2,720 sq. ft
Heating surface firebox	197 sq. ft.
Heating surface, total	2,917 sq. ft.
Steam pressure	185 lbs.
Weight, working order	198,000 lbs.
Weight on first drivers	43,000 lbs.
Weight on second drivers	43,800 lbs.
Weight on third drivers	45,900 lbs.
Weight on fourth drivers	43,800 lbs.
Weight on truck	21,500 lbs.
Weight of tender, loaded	104,600 lbs.
Water capacity	6,000 gal.
Coal capacity	22,000 lbs.
Least internal diameter of boiler	71 in.
Drivers, diameter	56 in.
Driving journals	9x13 in.
Engine wheel base	25 ft. 11½ in.

The weights of these engines are not given as absolutely accurate, but they are very nearly correct. The following figures were taken from H6 engine No. 321:

Weight:	
Truck wheels	20,131 lbs
Front driving wheels	44,000 lbs.
Intermediate driving wheels	40,200 lbs.
Main driving wheels	41,450 lbs.
Rear driving wheels	40,900 lbs.
Total	186,681 lbs.

These weights were taken with an average fire and about 2½ gauges of water in the boiler.

Boiler.

The boilers are of the Belpaire type, with curved crown and roof, and with the firebox above the frames, and with two cylindrical courses in front of the firebox. Cutting off the back head of H6 at an angle, besides lightening the back end and throwing the center of gravity farther forward, offers a more favorable arrangement of the firebox heating surface by removing the dead spaces at the upper back corners, and there is less radiation at long range while none of the available surface of the crown sheet is sacrificed. The space thus gained is used for an admirable arrangement of cab fittings. The sides of the boiler in the cab are flattened inwardly to further increase the room.

The sectional view of the boiler, Fig. 4, shows the water spaces to be 4 inches wide all around at the mud ring, and much wider above, where the necessity for long staybolts is greater. A free entrance for water is had at the throat through a space 5½ inches wide. Though there are 369 2-inch tubes, these are not placed near enough to the shell to interfere with free circulation, and a clear water space is left below them. The length of the tubes is 13 feet 7½ inches, and the diameter of the shell is 76 inches, the firebox is 10 feet long by 4 feet wide, the grate area being 33.3 square feet, and the heating surface 2,812 square feet.

The crown bolts are fitted with cup nuts, except the three bolts in each corner of the crown sheet, which are fitted with expansion joints. All of the stay bolts in the firebox, which are shown in Fig. 4 in the form of circles, are of the Nixon patent, illustrated in our issue of September, 1897, page 320. The following details of the boiler are interesting:

Fire area through tubes	6.39 sq. ft.
Grate area	33.33 sq. ft.
Heating surface tubes (external)	2,632 sq. ft.
Heating surface, firebox	180 sq. ft.
Heating surface, total	2,812 sq. ft.
Steam pressure	185 lbs.
Weight of boiler without flues	31,563 lbs.
Weight of flues	11,320 lbs.
Weight of boiler, total	42,883 lbs.

The outside sheets of the firebox are only ⅜-inch thick. This is unusual and is but half the thickness of the barrel sheets. The object was to favor the staybolts as much as possible and relieve them of the effects of the stiffness of thicker sheets. The presence of the stay bolts renders it possible to reduce the thickness, which in this type of firebox does not need to be greater than that of the inside sheets. This, the form of the firebox, the Nixon stays and the best staybolt material may be expected to give good results as regards staybolt breakages. The crown sheet is ⅜-inch and the flue sheets are ½-inch thick.

The mud ring corners are turned with 7-inch radii, which should help to prolong the life of the firebox. At the back end the sides of the firebox are straight and vertical, to give more room in the cab. Wash-out plugs are distributed as follows: 4 over the crown sheet, 2 in the top of the boiler, 2 (6-inch) under the boiler, 3 in the back head opening over the crown sheet and one at each corner of the legs of the firebox. The boiler has a dome, with a pressed steel top, using a straight sheet for the dome and cutting the flanged dome joint piece

also straight on top, as shown in the boiler drawing. The fire door is oval, which was the nearest possible approach to a rectagular hole, and at the same time avoiding the troublesome flanging of corners. The construction of these boilers merits an entire article. They are so carefully laid out and the work is so accurate as to be interchangeable. A back head was recently sent out from Altoona to be put into an H4 boiler at another shop, and, the work being done on formers and to templates, it was so accurate that it fitted in the boiler built some time previously.

Cylinders.

The cylinders are cast separate from the saddle, which at first sight may appear to be retrogressing, but there are several very good reasons why this is a good plan. It is not new except in application. The reasons are: (1) With this construction the shrinkage stresses, due to the form of the casting, which are so dangerous to the life of ordinary cylinders cast with the half saddle, are avoided. (2) The material for the saddle and the cylinders may be selected with reference to the work each has to do. A tough, strong iron, with comparatively low shrinkage, may be used for the saddle, and hard, close iron (which necessarily has high shrinkage) is used for the cylinders. There appears to be no difficulty to guard against the shrinkage when the castings are separated. (3) Separation simplifies the foundry problem, because it avoids the necessity for the chilling cores which were found necessary to spread the shrinkage and to prevent the formation of porous, spongy spots. (4) There is an advantage in repairs, because one cylinder may be taken down without removing the other cylinder or the saddle. (5) A two-bar frame is a necessary accompaniment of the cylinder and half saddle plan and with such an arrangement it is impossible to secure a frame connection as strong as the one before us. These reasons are probably responsible for the selection of this form of construction, and it is likely to be used in future designs in both freight and passenger locomotives on the Pennsylvania.

This arrangement of cylinders necessitates rigidity in the frame, the frame splice and the attachment of the cylinders and the saddles. The keying has been attended to with great care. The frames at the cylinders take the form of slabs 2 by 31 inches in section and vertical flanges on the inside and outside of each frame form contacts for the cylinders and saddle. The flanges at front and back are vertical and the wedges only are planed at an angle. The wedges are cut on the planer, together and by reversing pairs they must fit correctly. It would be impossible to secure a rigid fastening by attempting to plane the frames at an angle. This plan relieves the bolts from shear. They are in tension only. The saddle casting, which would otherwise have a tendency to rock in the frames, has four bearing surfaces at each frame, the center of the portion which meets the flange of the frame being cut away so as to clear the flange and remove the possibility of the saddle rocking in case the fit should be tighest at or near the center. This is one of the relatively small details that contribute to make this design so interesting and worthy of study. The strong ribbing of the cylinder connection plate is seen in Fig. 5. In the exhaust passage care has been taken to aid the steam in the direction of the nozzle by removing the corner or pocket usually found just below the exhaust port. This is shown in Fig. 6. The saddle is also very strongly ribbed and the steam and exhaust passages are separated with air spaces between them. Drainage of water from the two steam cavities at the ends of the cylinder casting is provided by means of a pipe leading to an independent cock under the center of the cylinder and operated by the cylinder cock lever. A further movement of the lever, after the cylinder cocks are open, opens this center cock to drain the steam cavities and the cylinder cocks close after this drainage cock is closed. With a leaking throttle this valve will do good service.

Frames.

With a maximum thrust of about 68,400 pounds from each cylinder, alternating as to direction, first on one side and then on the other, the frame problem is a difficult one and those who are having trouble with broken and loose cylinders will find these drawings valuable, because, so far, there has been no trouble of this kind. The front sections of the frames are cast steel and every precaution was taken to give rigidity and strength to the fastenings. The holding power is largely due to correctness of fit and arrangement of parts, which make this possible. The frames for both types are very nearly alike, Fig. 7. Aside from the transverse bracing afforded by the cylinders there is a cast iron bumper at the front ends of the frames, a cast steel plate across the frames at the frame splices and a strong connection brace at the back ends of the frames. The bracing against twisting stresses and those which come in taking curves on the road are provided back of the cylinders, where the tendency to wiggling actually takes place, because the designer believes that these stresses cannot be resisted by any amount of bracing placed in front of the cylinders. This is particularly applicable to the case of thin frames. The great vertical strength of this form, or slab frame, at the cylinders is also noteworthy.

The frame splice is interesting in that the keys are in compression and are relieved entirely from shearing stresses. This form has been used, as shown in the sketch, Fig. 8, which was adopted in 1892 for Class M engines. This is a very simple and important improvement. It is so easy to see its advantages, in avoiding shear and in the possibilities of securing a good fit, that one wonders that it has not been used always. The splice of Class H6 is an improvement upon the earlier form in the number of bolts used.

The expansion pads and frame clamps are shown in Fig. 9, the rear clamps and pads being similar to the front ones, which are illustrated. The parts are arranged to relieve the studs of the usual tearing stresses. The firebox tends to push against the studs, as will be easily seen in the engraving. No wear is permitted on the mud ring or the frames. The supports for the frames and also the spring seats are provided in the steel castings fitted between the upper and lower bars of the frames.

Running Gear.

Pistons.—The pistons, Fig. 10, are made with a steel centre in the form of a dished single plate to which is bolted a T shaped ring of cast iron with two packing rings. The thickness of the piston for a distance determined by an angle of 60 degrees from each side of the center at the bottom is 5 inches, for the purpose of increasing the bearing surface, while for the remaining 240 degrees the thickness is reduced to 3⅜ inches. This increases the wearing surface where it is needed, without unduly increasing the weight of the reciprocating parts. The cylinder heads provide for the thickening of the pistons without increasing the steam clearance space. The entire surfaces of the pistons are dressed.

Piston Rods.—There is nothing unusual about the piston end of the piston rod. It is drawn up to a shoulder by a nut, but the crosshead end is exceedingly interesting, because there has been no piston rod breakage with these engines and with this design it is safe to say that there will be none unless due to bad fitting. The piston rod is shown in Fig. 11, and the cross head in Fig. 12. The rod bottoms in the cavity in the crosshead and the key does not stretch the rod (see "American Engineer," January, 1899, p. 9), but this is not all. The taper is increased to prevent too hard driving and the taper portion of the rod is cut away at the central part of its length, leaving bearing surfaces 2 3/16 inches long at each end. This is the idea followed on the cylinder and saddle joints and in the bearings of the rocker shafts. In the case of the crosshead fits it prevents the rocking of this connection, which results from swelling the piston rod by cutting the key way. A great aid in good fitting is provided in the slope of the key way in the piston rod and crosshead boss, as shown in the piston rod drawing, Fig. 11, the key slot being at right angles with one of the sides of the tapered end of the rod. The bearing against

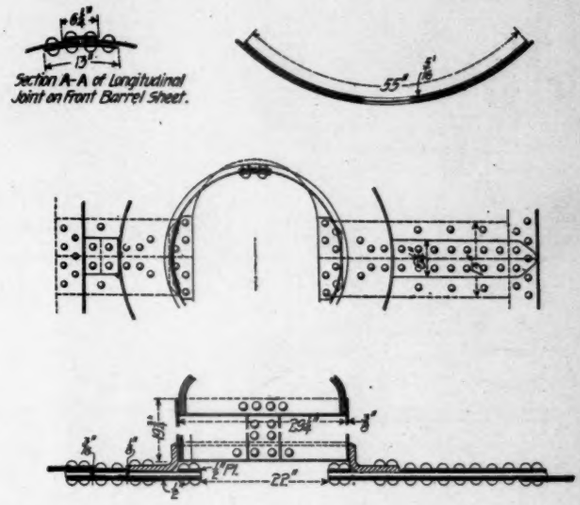
the crosshead boss is square. The end of the rod is, of course, enlarged.

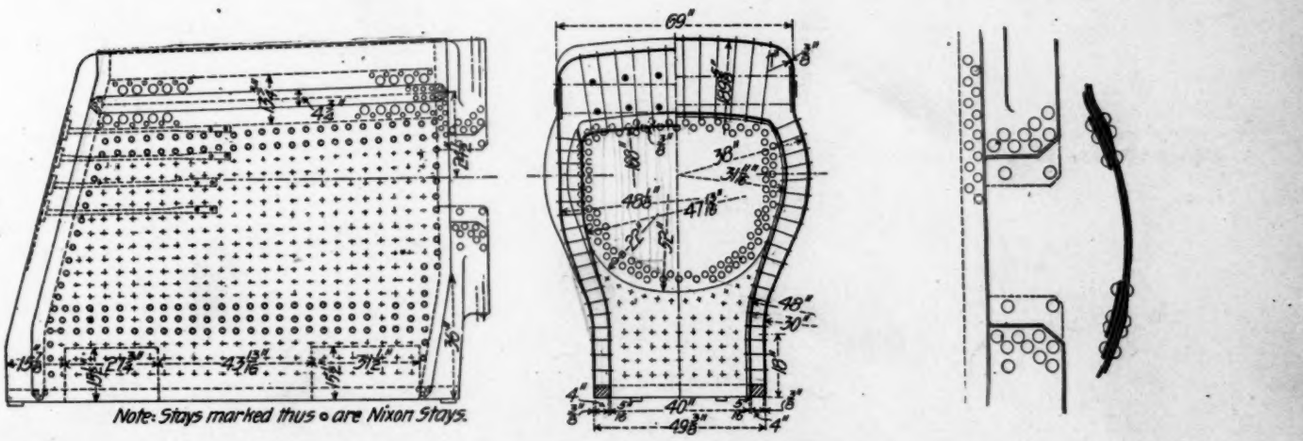
Guides and Crossheads.—The guides, Fig. 13, are of the two-bar type and of steel and 6 inches wide, with raised wearing surfaces on the sides in order that when truing up it shall not be necessary to also plane off the guide lugs. The crossheads, Fig. 12, are cast steel with block tin wearing surfaces 1/16 inch thick and with 2-inch wearing surfaces on the edges of the guides. The guide shoes are secured to the crossheads by lock nutted bolts. This is a very light crosshead for such a heavy engine.

Driving Boxes.—The driving boxes, Fig. 17, are not symmetrical with reference to the journal bearing. The total length of the bearing is 13 inches, and the center of load is 1 inch outside of the center of the box. The object of this is to throw less weight upon the inner edge of the box, where the greatest wear generally occurs. The driving boxes are still more interesting in their methods of lubrication. The bearings are phosphor bronze and they have no openings at the top of the journal, but grooves ⅜ inch wide, located 1¼ inches above the center of the axle and extending across the box, front and back, to within ½ inch of the ends of the bearing. The reason for this method of introducing the oil will be found in our March, 1898, issue, page 91. The oil is fed from a cavity in the top of the driving box and the best thing about this idea is that it is successful and the journals very rarely heat. The hubs of the wheels are lubricated by grooves and the oil for this purpose is held in a separate cavity in the top of the box. The large cavity is for the journal. The wedges and shoes are oiled by four grooves at each side, receiving oil from other cavities at the front and back sides of the top of the box.

Crank Pins.—The enlarged wheel fits and ample fillets of the crank pins appear in Fig. 14.

Driving Axles.—The main axle, Fig. 16, shows the principles of construction of all the driving axles, which are of steel. The journals are large and the wheel fits are one inch larger than the journals in diameter. We desire to direct special attention to the form of the key ways in these axles. They are made with cutters, having a radius of 2¾ inches. Also, the enlarged ends of the axles where they enter the wheels are worth imitating. A very slight shoulder is left against which to bring the wheel hubs in pressing them on.





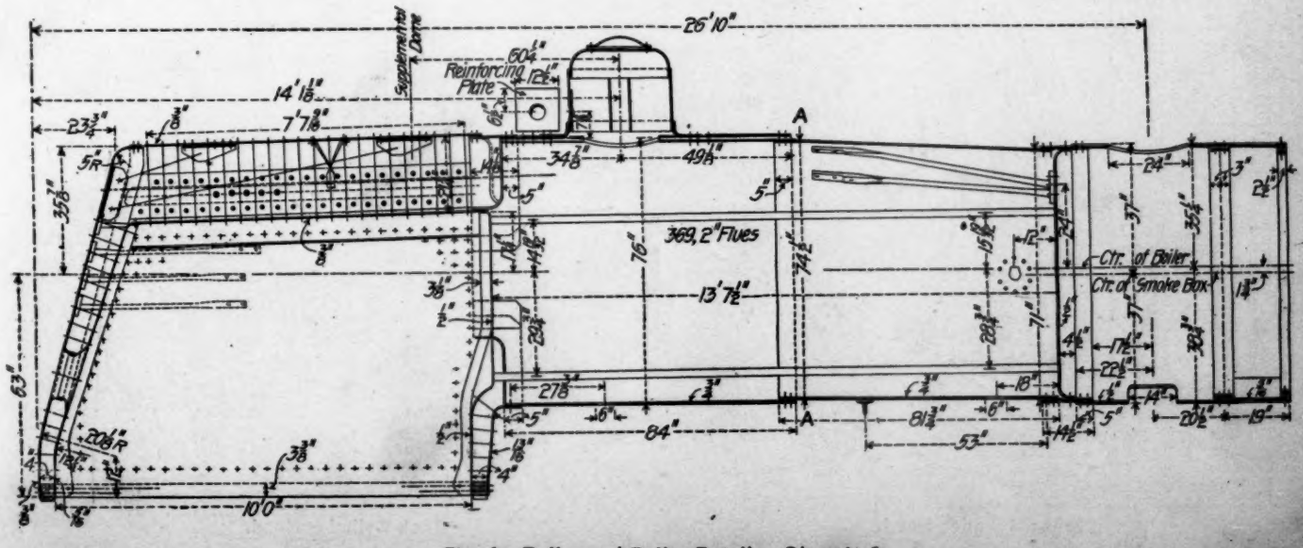
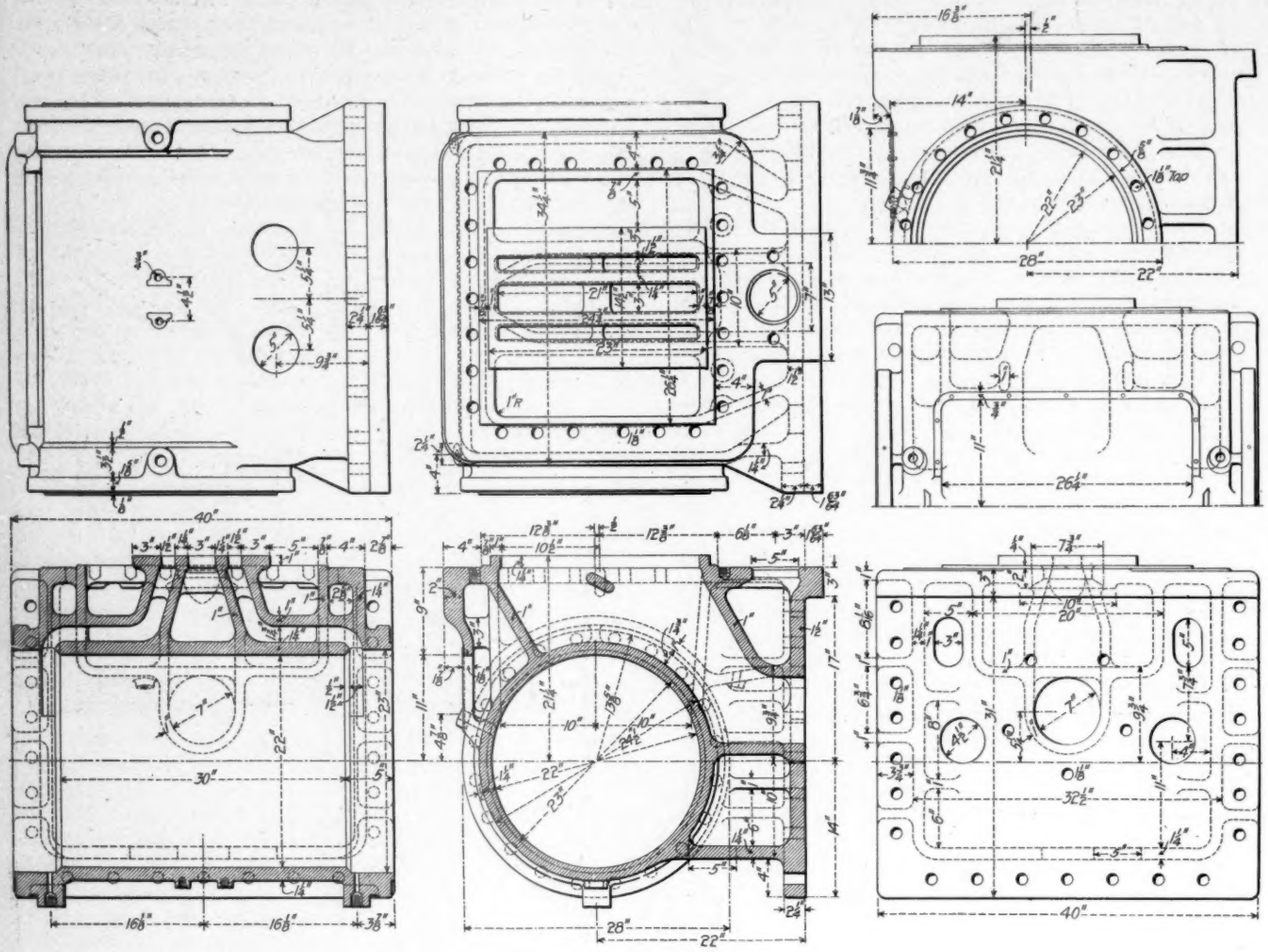
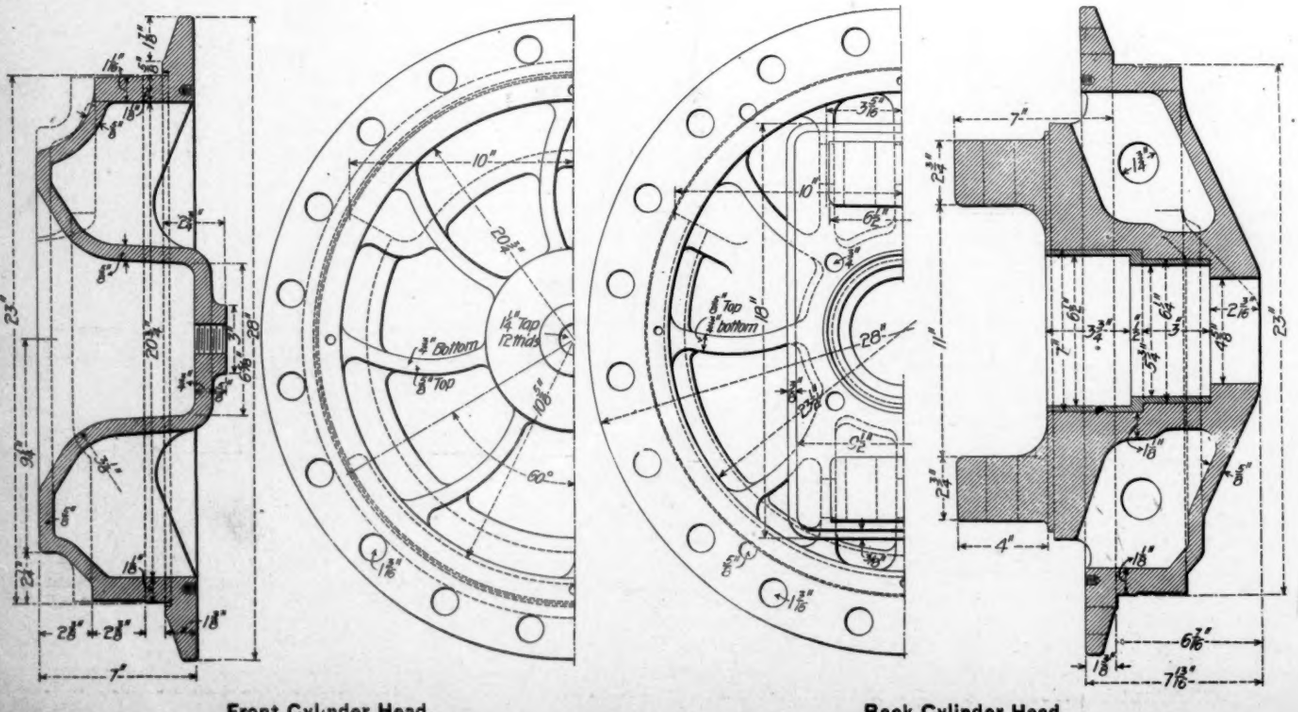


Fig. 4.—Boiler and Boiler Details.—Class H 6.



Cylinder and Cylinder Heads.



Front Cylinder Head. Back Cylinder Head.

FIG. 6.—CYLINDER AND CYLINDER HEADS, CLASS H6.

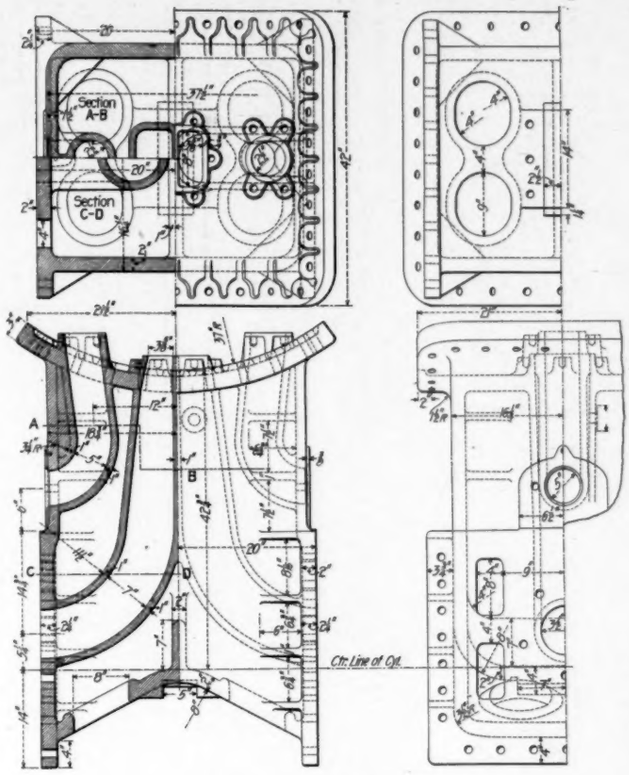


Fig. 5.—Saddle.

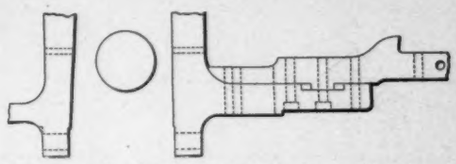


Fig. 8.—Frame Splice, Class M. Engines, Used in 1892.

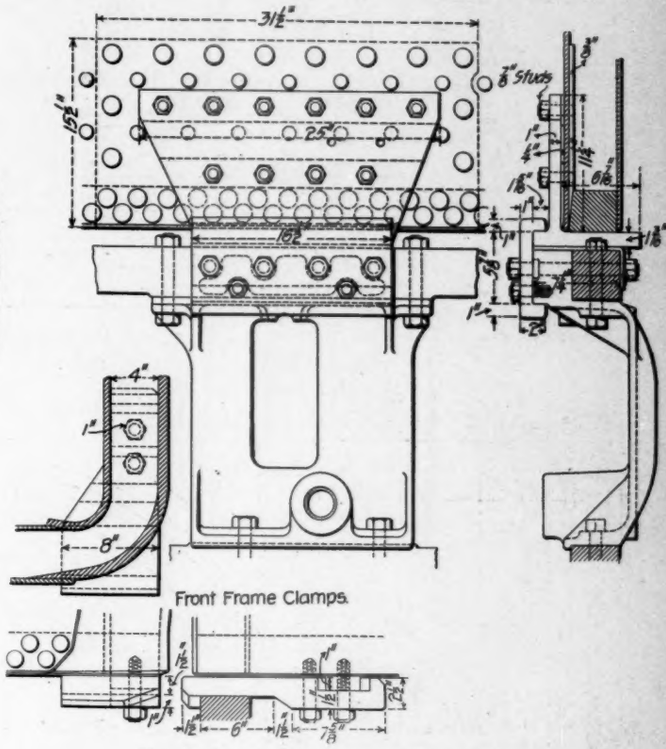


Fig. 9.—Expansion Pad and Frame Clamp.

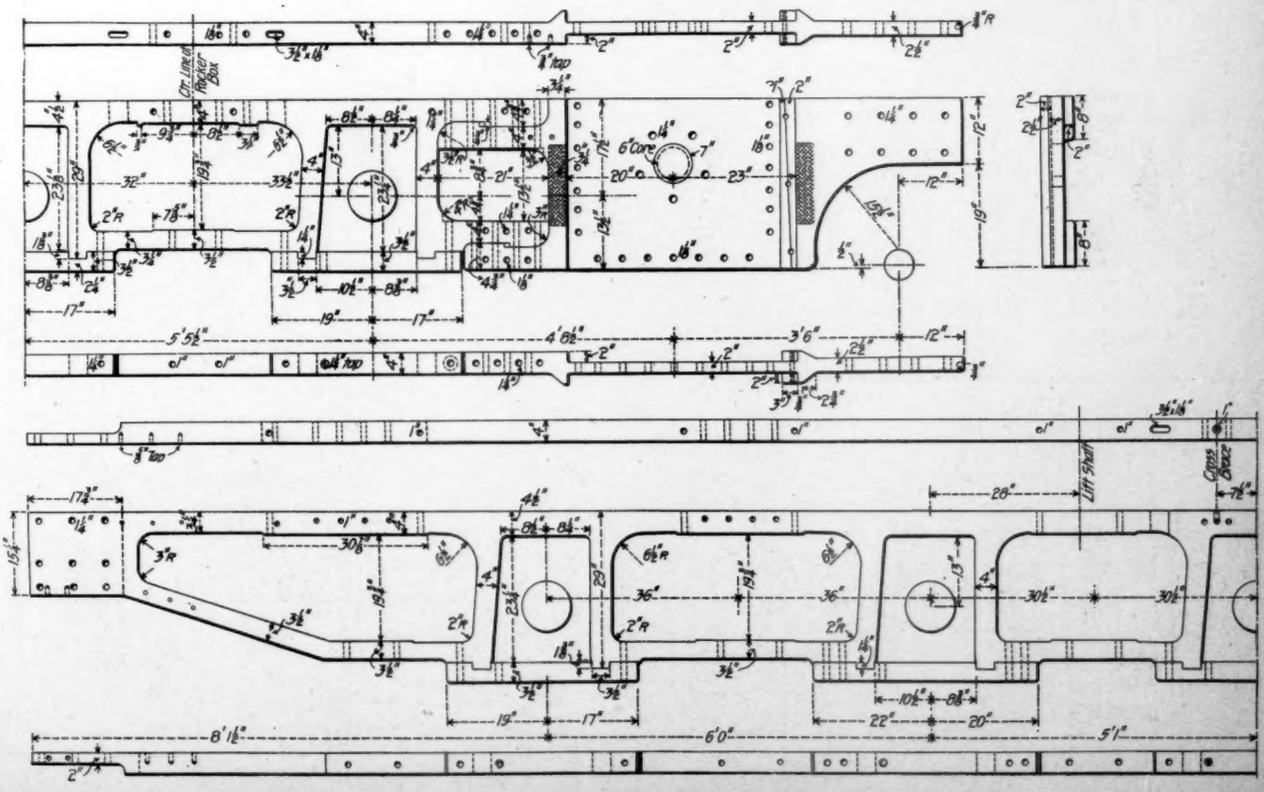


Fig. 7.—Frames—Front Portion of Cast Steel.

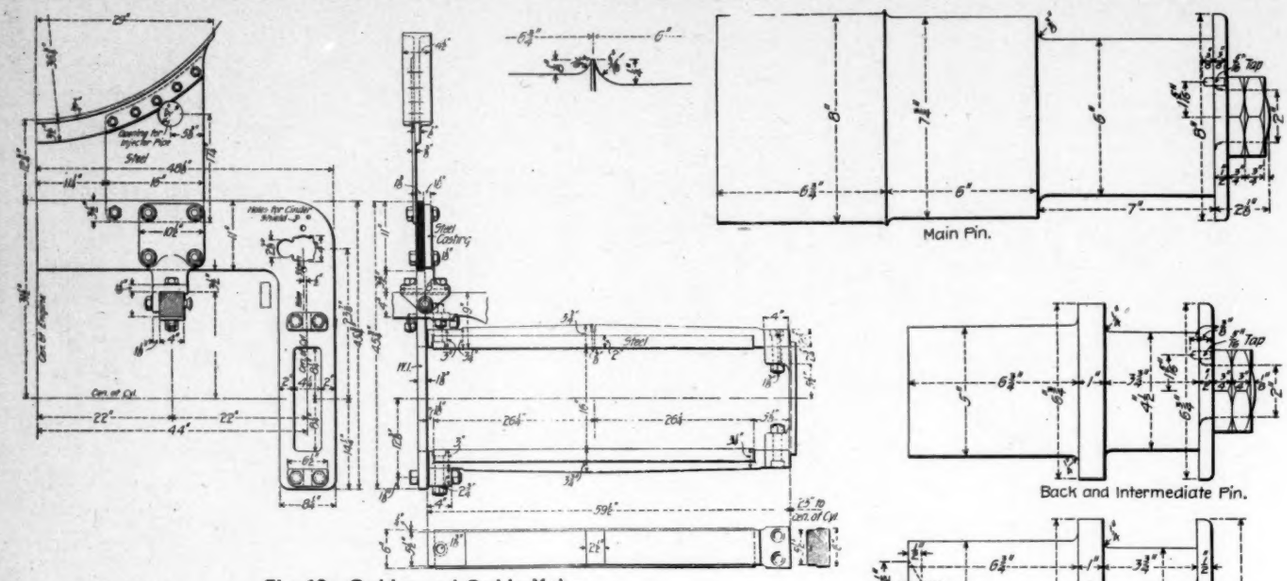


Fig. 13.—Guides and Guide Yoke.

Fig. 14.—Crank Pins.

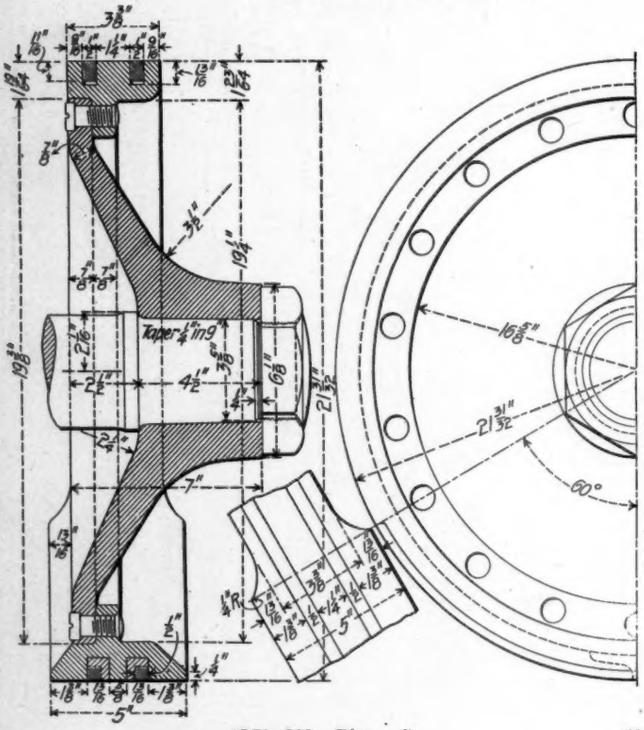


Fig. 10.—Piston.

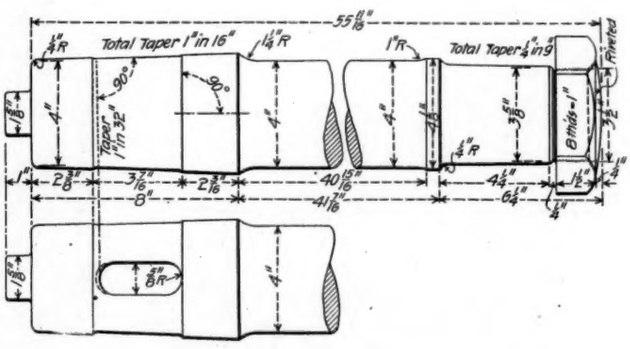


Fig. 11.—Piston Rod.

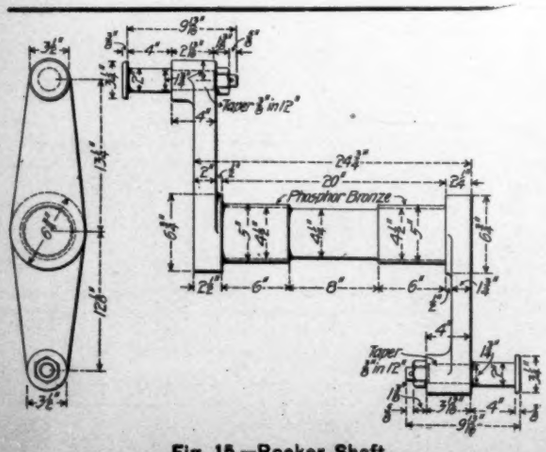


Fig. 15.—Rocker Shaft.

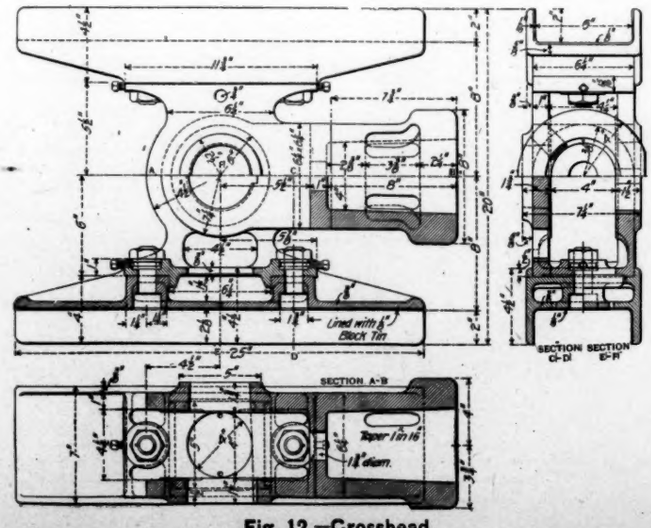


Fig. 12.—Crosshead.

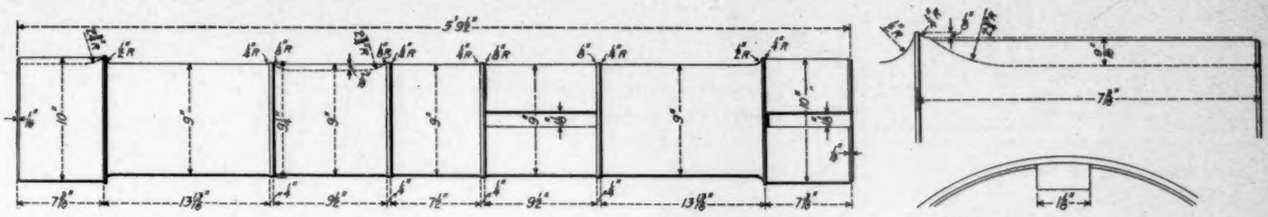


Fig. 16.—Driving Axle.

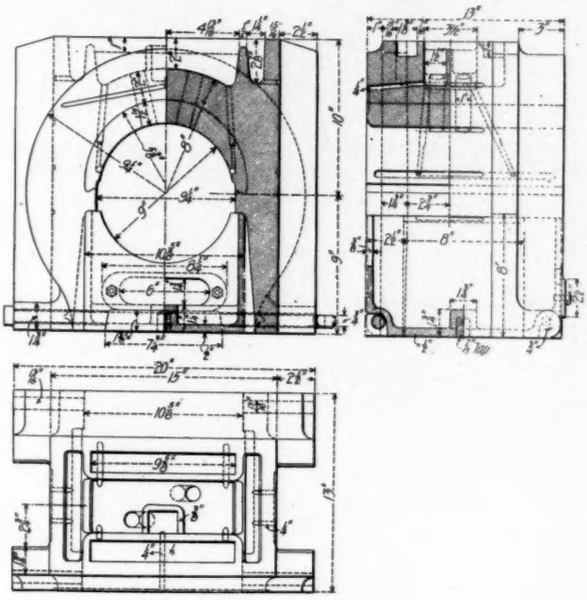


Fig. 17.—Driving Box.

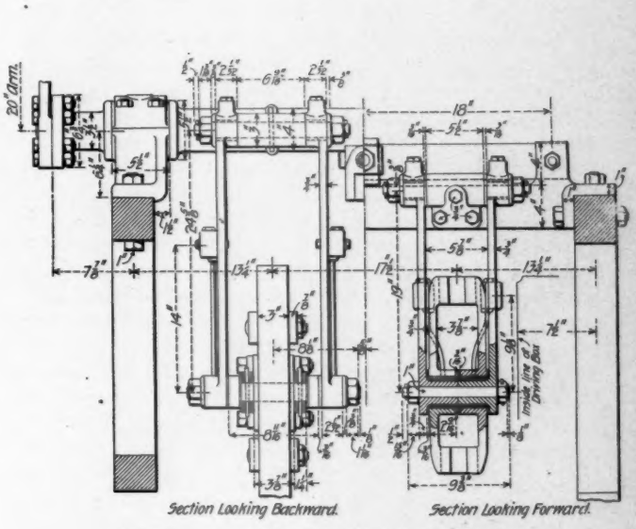


Fig. 18a.—Sections Through Valve Motion.

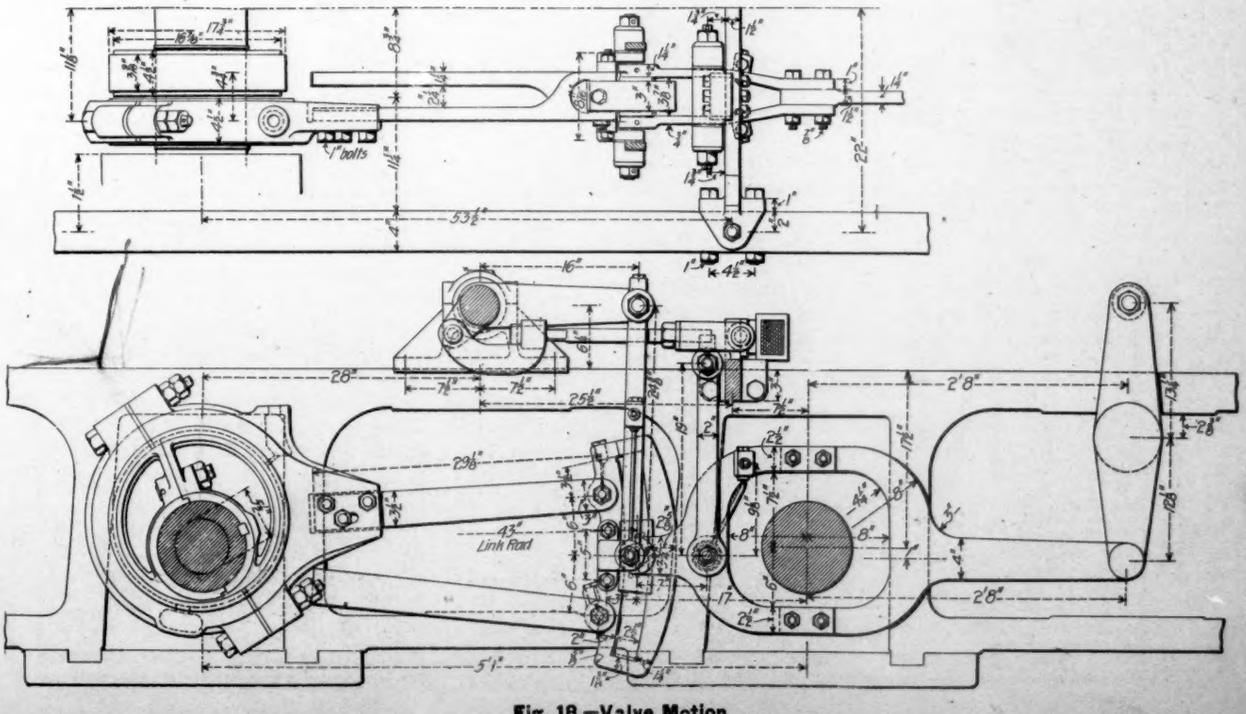


Fig. 18.—Valve Motion

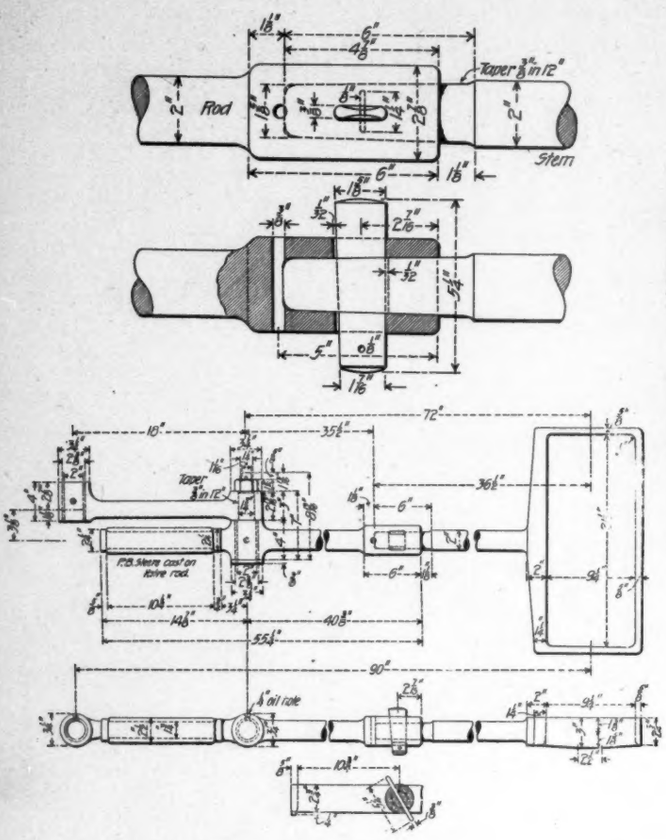


Fig. 19.—Valve Stem and Yoke.

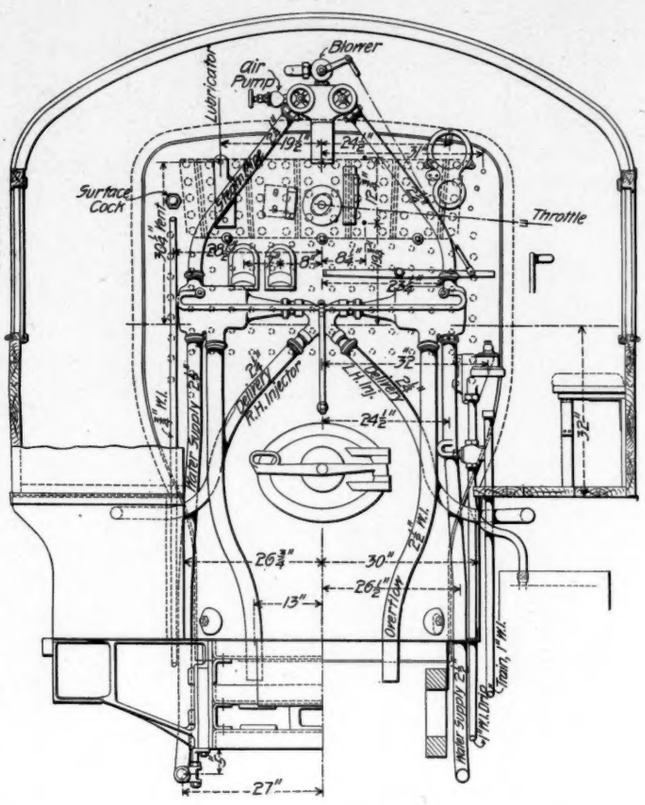


Fig. 20.—Cab Fittings.

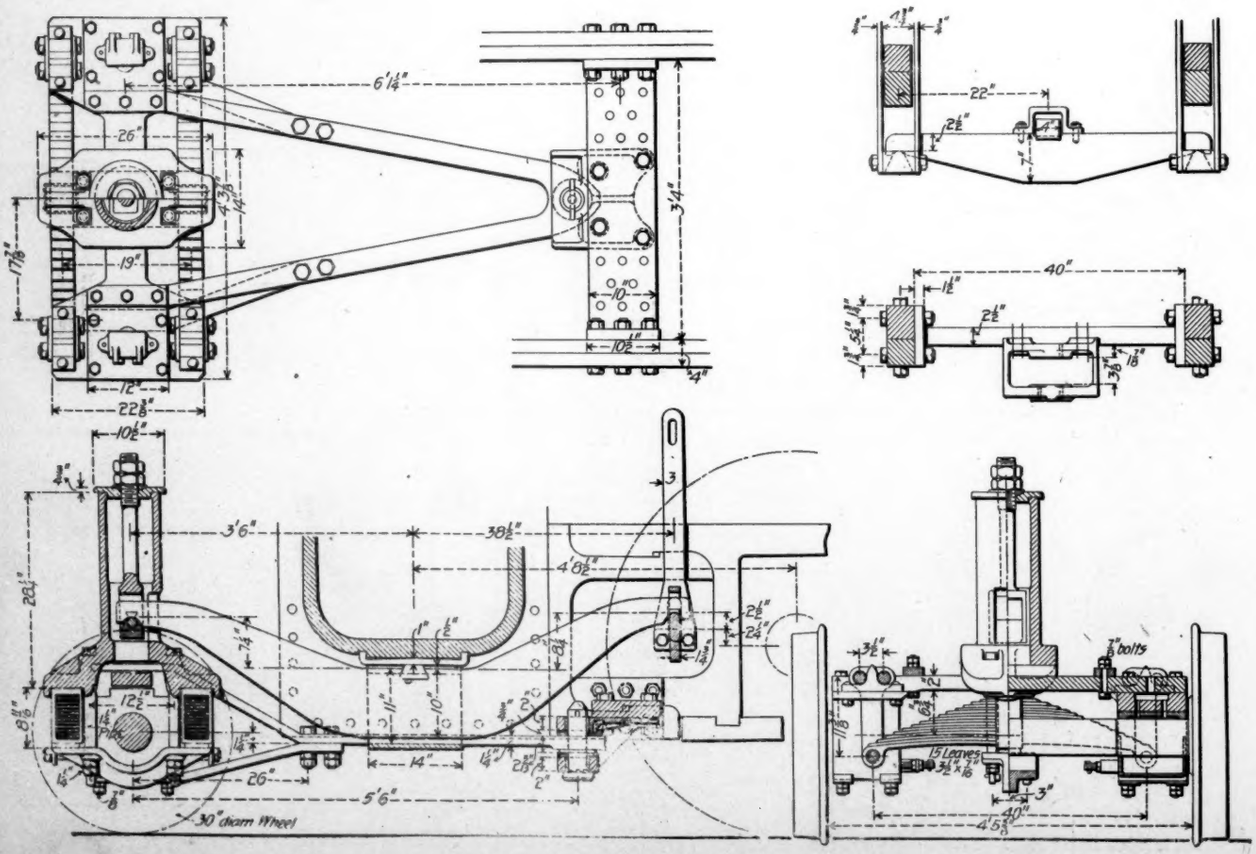


Fig. 21.—Engine Truck and Suspension.

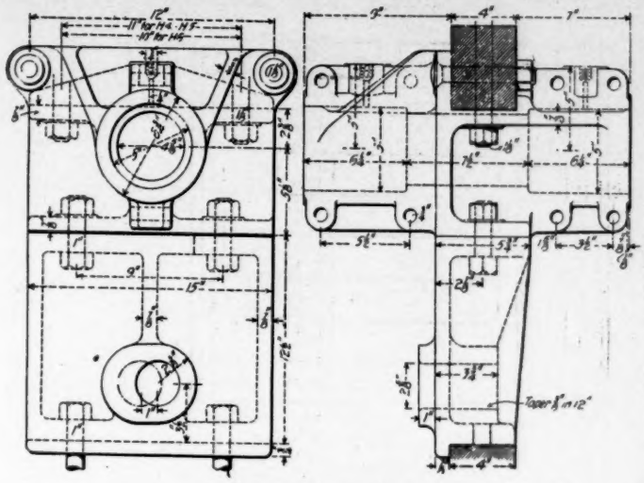


Fig. 22.—Rocker Box.

Driving Wheels.—Cast steel was used for the driving wheels, not altogether because of the saving in weight, but because the wheels are thin, being but 7 inches over the hubs, and it was decided that no chances of breakage should be taken. The front and rear driving wheels only are flanged.

Valve Motion.

The rocker box is between the first and second pairs of driving wheels, which makes it necessary for the valve gear to pass the second axle. This is done by means of a closed loop, shown in Fig. 18. From the joint at the axle the loop is integral with a 1¼ by 4-inch rod connecting to the lower rocker arm, while the connection from this joint to the link is double and is best seen in the plan views of Fig. 18. The weight of the back end of this connection is carried by double links supported on a cross piece between the frames, Fig. 18a. The attachments to the link block and to the link hangers are double and such as to transmit a straight line pull on this connecting piece, to the relief of the links and eccentrics from a twisting tendency. The drawings show the construction of this interesting detail. This valve motion is the most direct and rigid that we have seen, particularly for one which spans the driving axle. The motion is easily disconnected and by use of reamed bolt holes there is no possibility of making changes. Careful arrangements have been made for oiling. The link is strong and is reinforced by additional material at the center portion. The radius of the link is 43 inches, but the distance between centers is made 44 inches, for the purpose of reducing the lead at short cut-offs. The angle of the lifter spring arm is so taken as to decrease the lever arm as the spring compresses, which gives nearly uniform resistance.

The rocker boxes, Fig. 22, for Classes H4, H5 and H6, are alike. They are below the frames and to their lower faces filling pieces are fitted which bear on and are secured to the lower bars and form pivots for the brake hangers. The idea of preventing rocking of a bearing by the possibility of its being tightest in the middle of its length is carried out here by turning down the central portion of the rocker shaft and causing it to bear in the rocker box only for a length of 6 inches at the ends. These bearing surfaces are of phosphor bronze in the form of sleeves, ¼ inch thick, cast upon the shafts, as shown in Fig 15. The valve stems and yokes are shown in Fig. 19, in which the knuckle joint and the phosphor bronze sleeve cast on the back end of the valve stem are shown. This sleeve has a bearing that is bolted to the guide yoke and over it are cinder shields of cast iron.

Engine Truck.

The truck is the result of wide experience, which has shown methods by which repairs may be reduced. It is illustrated in Fig. 21. The truck boxes are bolted on each end, to a straight backbone, with a palm on each end, causing the boxes to wear uniformly. The backbone carries no load and does not cause the boxes to tilt relative to the journals. The weight is carried by initial stability links, which have obvious advantages over the common form of swinging links, and the load is transmitted to them by two springs with 15 3½ by 7/16 inch leaves. A spring keeper, under the axle, will catch the springs in case of failure of the hangers and prevent them from falling on the track. A 2½ by 10-inch brace extending across the engine at the frame splices and with broad flanges at the ends bolted between the frames carries the pivot for the radius bar. This brace has six bolts at each end and serves also to stiffen the frames laterally. The radius bar is low and its center is 1¼ inches below the center of the axle. It has a forked attachment to the truck and the stresses and shocks are provided for in such a way as to relieve the braces from a large part of the usual shearing effects. The equalizer passes below instead of through the saddle. The boxes are accessible and the pedestal caps are not removed in order to get at the sponge boxes. These are held in place by wedges that are easily taken out. A heavy cast iron spring guard at the center of the axle is used for the purpose of catching a broken spring. This is the old Pennsylvania truck simplified and designed to make use of castings instead of expensive forgings.

Lesser Details.

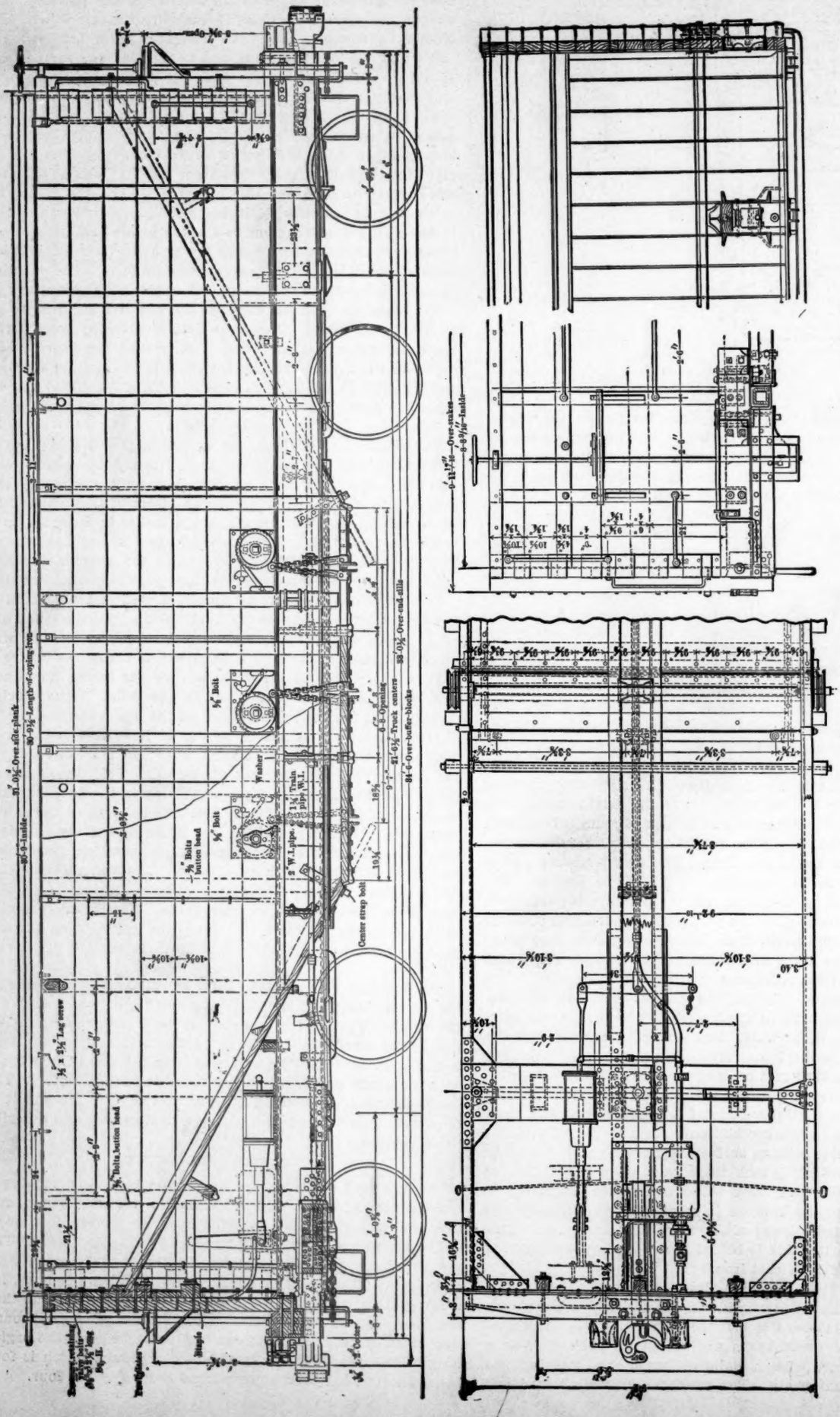
The smoke box is short, the extension proper being but 16 inches long and the whole smoke box being 66½ inches long. The stack is of cast iron, extended inside the smoke box 28¾ inches with a bell at the bottom of the extension and the tip of the nozzle, which is single and 5½ inches in diameter, is 20 inches below the bell. The stack and base are in one casting. The blower is in the form of a ring and the air pump exhaust is brought into the main exhaust passage in the saddle casting.

The air brake cylinders of Class H6, Fig. 3, are secured to the frame brace immediately back of the cylinders and the brake shoes are back of the wheels. One object of this was, like the sloping of the back boiler head, to throw the center of gravity ahead. There are other advantages in the horizontal pull of the air brake piston and in the relief of the driving springs from the downward thrust of the brake shoes.

The cab fittings as shown in Fig 20 have received a great deal of attention. Figs. 1, 2 and 3 show the stop valve in the cab supply pipe just in front of the cab. This, when closed, permits of disconnecting any steamfitting in the cab, with the exception of the steam gauge. The arrangement of the cab is admirable. Even the whistle is a piece of excellent designing. It is mounted on a connecting piece, which attaches to the dome, and in it is a stop valve. The whistle operating valve is back of the whistle and cannot drop into the steam pipe when disconnected for regrinding. The lever is double with the spring between the two parts. Mr. Vogt's throttle with double fulcrum lever is fitted to these engines.

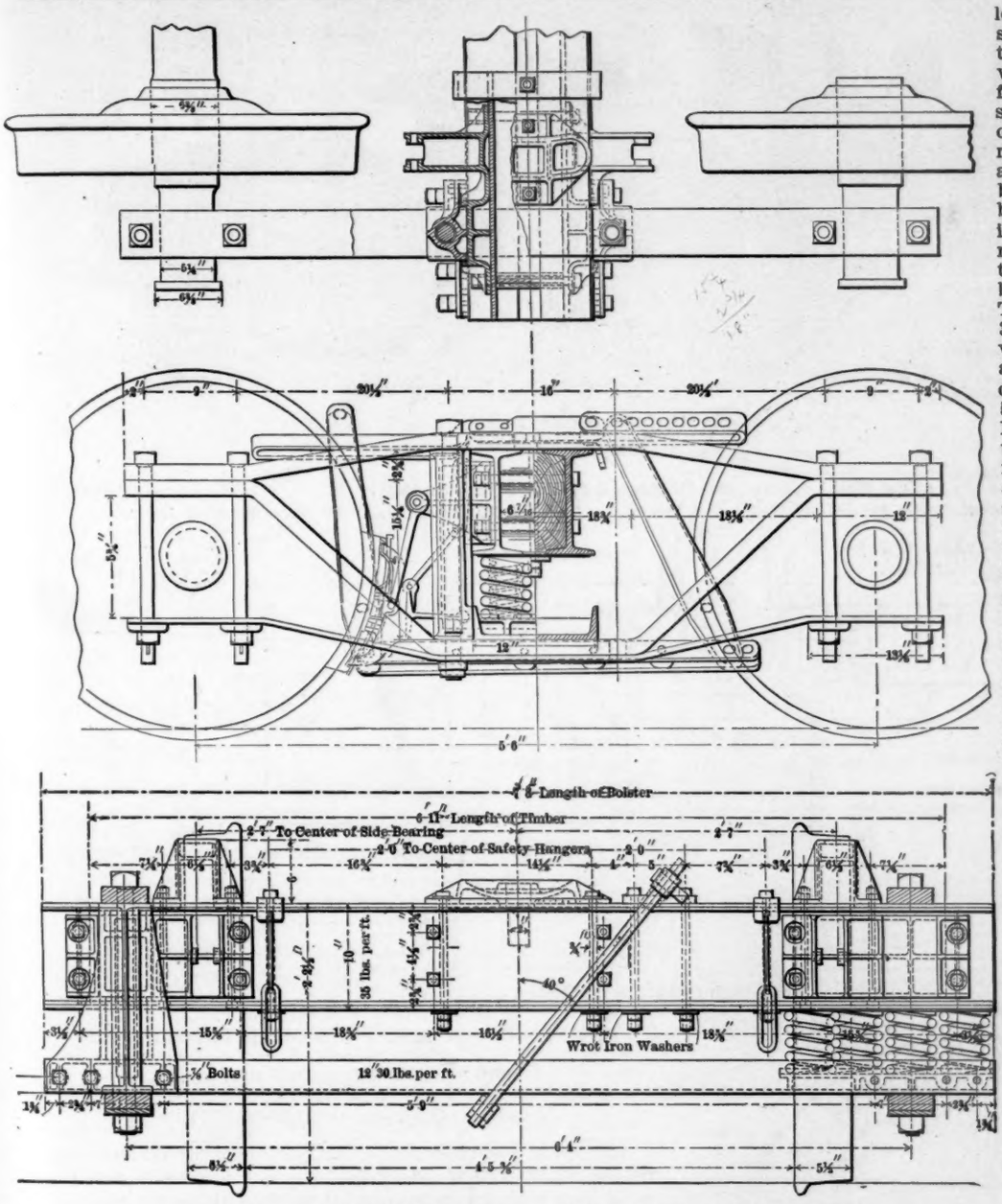
It is impossible to examine these locomotives without admiring the excellent and thorough engineering of which they are the result. They are conservatively bold. The conservatism lies in the careful investigation of the merits of each feature before it is used and the boldness is in the use of new ideas without hesitation when they are found to be satisfactory. The workmanship is in every way equal to the planning and it is altogether a privilege and a pleasure to illustrate and describe these engines.

The fast mail trains on the Burlington have been making a number of fast runs. April 23 train No. 15, west bound, ran from Chicago to Burlington, 205.8 miles, in 199 minutes, or at at the average rate, including stops, of 62 miles per hour. The portion of the run from Clyde to Burlington, 197.3 miles, was done in 184 minutes, or at the rate of 64.33 miles per hour, including stops. These are not record breaking runs, but they are very fast for regular service conditions. The east bound mail train on this road has been late but once since the beginning of this year, and the west bound schedule, which is four hours shorter, has been accomplished every day but four.



Steel Frame Hopper Cars, 100,000 Pounds Capacity.
Norfolk & Western Railway.

W. H. LEWIS, *Superintendent of Motive Power.*



Truck for 100,000 Pound Cars.—Norfolk & Western Railway.

100,000-POUND, STEEL FRAME, HOPPER CARS.

Norfolk & Western Railway.

On another page of this issue Mr. R. P. C. Sanderson, in describing the present status of the steel car, alludes to a new design by the mechanical department of the Norfolk & Western Railway which, through the courtesy of Mr. W. H. Lewis, Superintendent of Motive Power of the road, we are permitted to illustrate and describe. In connection with the description and engravings, Mr. Sanderson's arguments favoring this form of construction should be considered, and it will also be interesting to make comparisons between this design and that of the Northern Pacific Railway, illustrated in our issue of October, 1898, page 334.

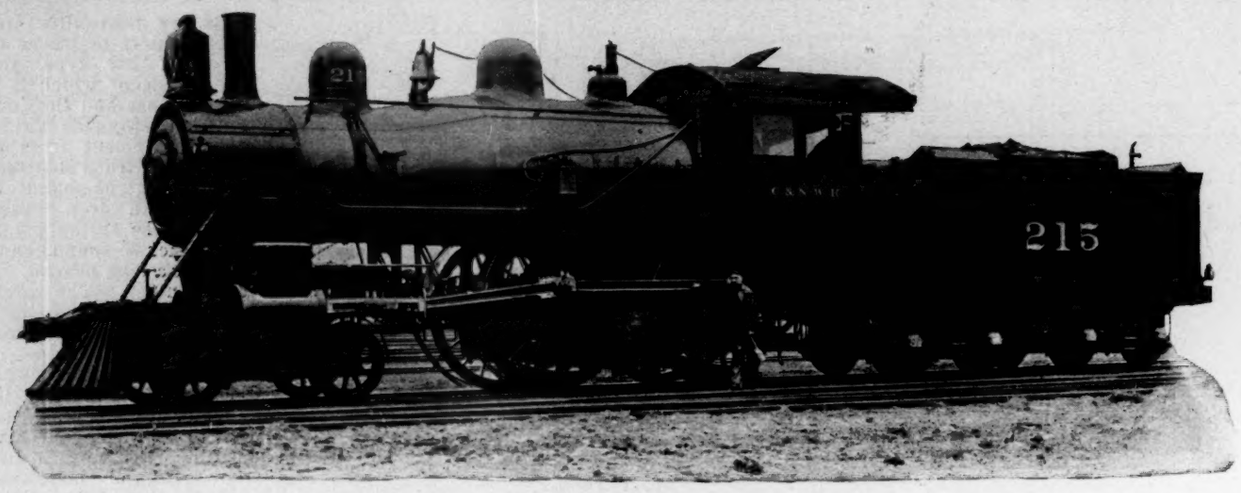
Like the Northern Pacific cars, the Norfolk & Western design has wooden hoppers. The underframing of the car consists of four 15-inch 33-pound channels connected with plate transoms and end sills. Except the crossties belonging to the construction of the hoppers, there are no cross braces, the center sills are rigidly secured together with iron space blocks, which serve also for suporting the train pipes and brake rods. The construction of the connections and the bracing of the frames is clearly indicated in the drawings and requires no extended explanation. The draft gear brings its stresses directly upon the center sills through tandem springs, both of which are in action, whether the car is being pulled or pushed. The underframe is entirely of steel, the hoppers being of wood and the loads are transmitted by steel bolsters to trucks of the arch bar type, the wheel base of which is 5 feet 6 inches and the construction being standard in other equipment of this road. The truck bolsters are 10-inch 35-pound I-beams with iron space blocks. The spring plank is a 12-inch 30-pound channel with a long bearing on the columns. The arch bars are 1½ by 4½ inches. The wheels are of cast iron 33 inches in diameter, weighing 650 pounds. The axles are 5⅜ inches in diameter at the center with 5¼ by 9-inch journals. Most of the castings in the trucks are of malleable iron and the center plates are of cast steel. The M. C. B. standards for 5 by 9-inch journals are used in the journal box, the journal bearing and wedge except as to the diameter of the journal. The cars are not to be used in interchange service and in view of the fact that no standard has been established for 100,000-pound axles and journals, the designers felt free to use such proportions in the axles and journals as appeared to be justified by experience.

The wooden hoppers have drop bottoms of the type used by this road in other equipment, except that in this case three doors are employed, the construction of which is clearly indicated in the drawings. The chains for handling the doors have been kept upon the outside of the car for the purpose of avoiding damage while unloading the cars. They are arranged to be operated from either side and the chain sheaves are mounted on heavy shafts extending across the car, these being protected by 2½-inch pipe, which also serves to tie the sides of the hoppers together. The ties at the bottom are in the form of cross pieces, to which the doors are attached. Four one-inch rods serve as top ties and these are provided with thrust timbers protected by angle irons. The side and end boards of the hoppers are 2¾ inches thick, experience having demonstrated the necessity for this precaution. The bottoms of the hoppers are of 1¾-inch oak. The sides are secured by stakes carefully proportioned to prevent bulging and thoroughly secured to the sills and sides of the hoppers.

The design has been very carefully studied and a thorough examination of it conveys the impression of simple strong construction intended for continuous severe service under exacting traffic conditions. The specially noteworthy feature aside from the form and arrangement of the underframe is the employment of commercial shapes of structural steel and the absence of special or unusual sizes of material, the purpose being to facilitate repairs from stocks that are ordinarily available.

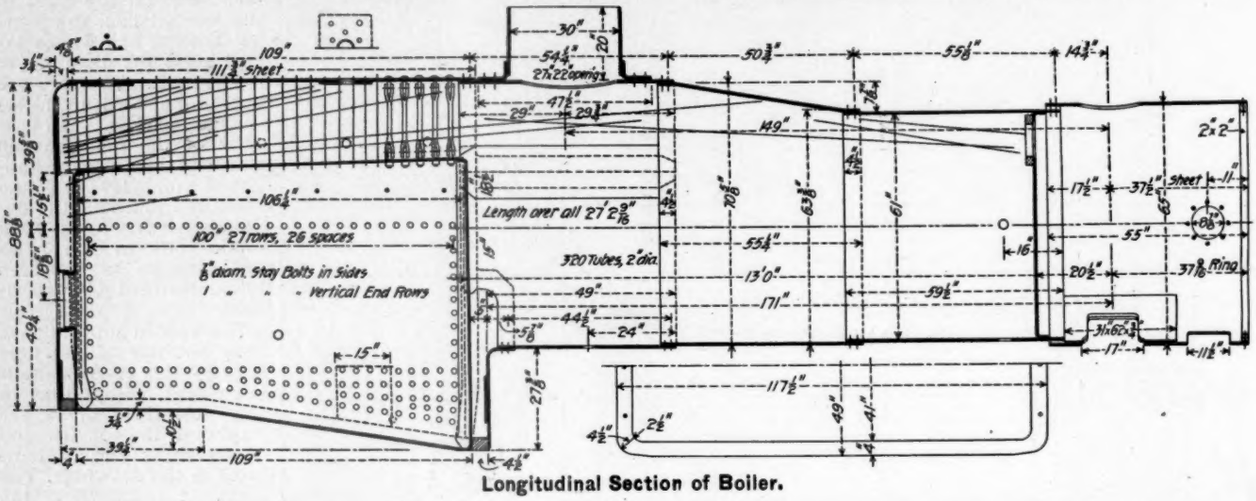
The weight of the car empty will be about 38,000 pounds, but the exact weight is not yet known. A recent test with a sample car loaded with coal showed a deflection of less than ¼ inch on all four of the sills, which indicates that the fibre stresses in these members have been kept below the calculated amount, which was 12,000 pounds per square inch. Orders have been placed for material for building 1,000 of these cars, the work to be done at the Roanoke shops of the road.

Mr. Sanderson's article will be found on page 190.

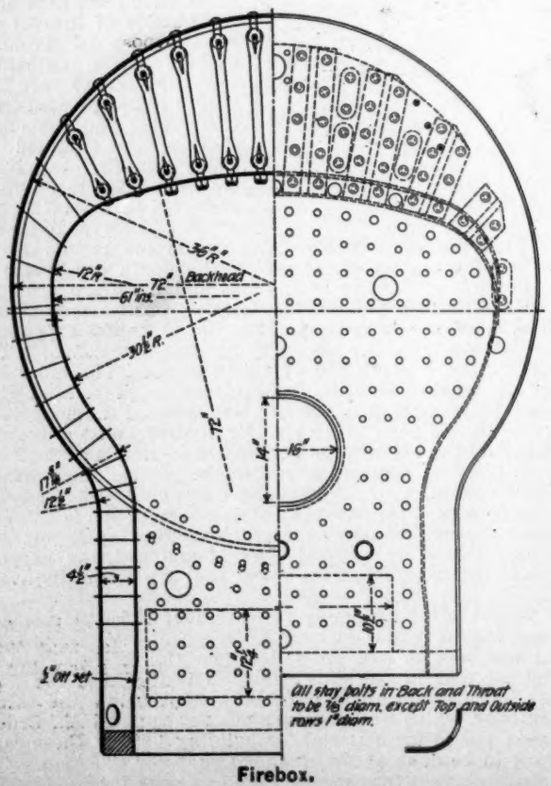


Fast Mail Locomotives—Chicago & Northwestern Railway.

ROBERT QUAYLE, *Superintendent Motive Power.* W. H. MARSHALL, *Assistant Superintendent Motive Power.*



Longitudinal Section of Boiler.



Firebox.

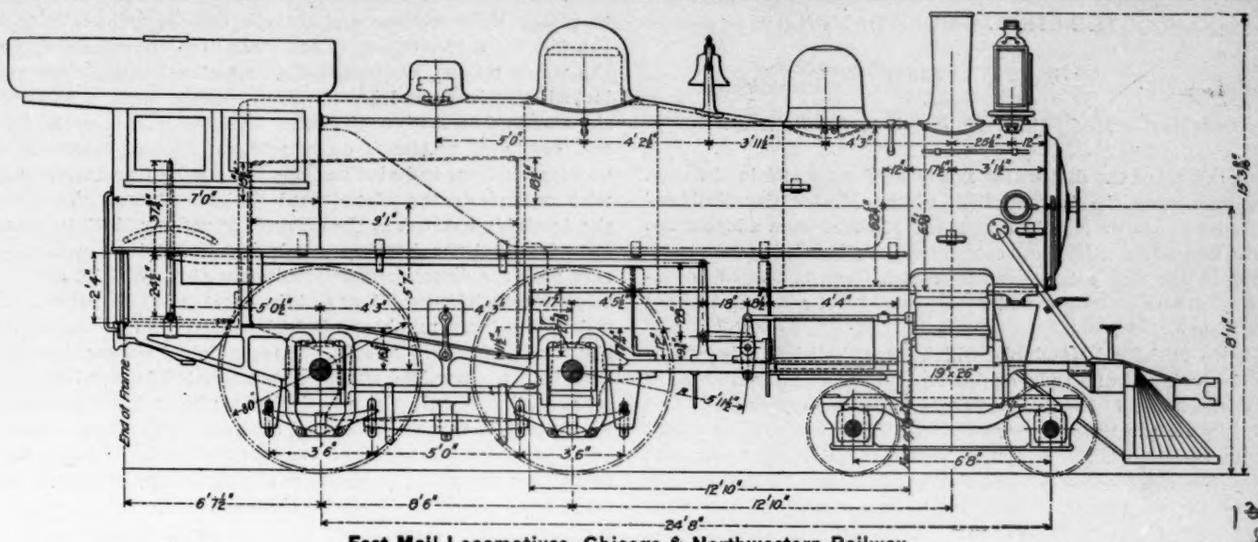
POWERFUL 8-WHEEL PASSENGER LOCOMOTIVES.

Chicago & Northwestern Railway.

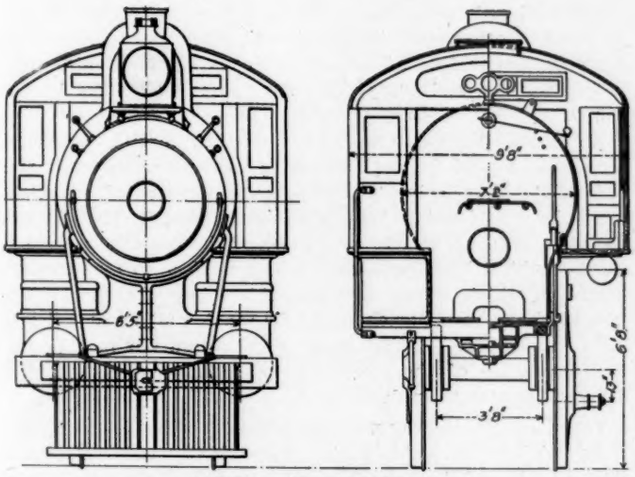
The Most Complete Description Published.

The increase in the power of fast passenger locomotives since the appearance of the New York Central No. 999 in 1893 has been very marked. Our previous issue contained on page 141 a description of the heaviest engine built for this service. The tendency has been toward the Atlantic and ten-wheel types for the heaviest engines, and the two recent designs by the Chicago & Northwestern, built by the Schenectady Locomotive Works, are of unusual interest because of the great power which has been obtained with the eight-wheel type. The question of type is exceedingly important and the Northwestern locomotives are remarkable in that they very nearly approach the maximum capacity in passenger service without resorting to additional wheels. They are the result of efforts to secure the best advantages to be obtained with 8 wheels, and the efforts are apparently successful to a remarkable degree. They represent the best designing we have seen for simple eight-wheel locomotives. To attract attention to these engines it is enough to say that 2,514 square feet of heating surface is carried on 8 wheels. This has never been done before and this heating surface has not been exceeded by that of any passenger locomotives. This heating surface is obtained without excessive length of tubes, and in this connection it is interesting to note that the Burlington engines have tubes 16 feet long, which make the heating surface 2,510 square feet.

The two C. & N. W. designs are for the fast mail and for heavy passenger service. The fast mail engines have 19 by 26-



Fast Mail Locomotives—Chicago & Northwestern Railway.



End Elevations and Sections.

inch cylinders, 80-inch driving wheels, 62-inch boilers, 2,364 square feet of heating surface and weigh 133,800 lbs., of which 85,700 lbs. are on the driving wheels (within 50 lbs. of the corresponding weight of the Burlington, Atlantic type engines). The heavy passenger engines have 19½ by 26-inch cylinders, 64-inch boilers, 2,514 square feet of heating surface and 75-inch driving wheels.

These engines have no novel features. They represent the best of present practice and the chief interest in the details lies in the ample bearing surfaces and in the elimination of unnecessary weight in order to favor the boilers to the utmost. Driving journals 9 by 11½ inches, and truck journals 6 by 12 inches, represent the strength and area of the bearings, and an examination of the driving wheels indicates how the weights have been kept down. The smoke box has a pressed steel front and door, the rocker boxes are cored out, the cab brackets are of ¼-inch steel plate. The cast steel parts are as follows: Driving wheels, driving boxes, eccentric straps, foot plates, expansion plates and brackets, rockers, cross-heads, front bumper knees, couplers and tender draft castings. The fire door is of pressed steel and this saves about 200 lbs. in material. The plate cab brackets save about 300 lbs.

To the information given in the table the following may be added. These engines have the high speed brake, brakes on the truck wheels, Leach's improved sanders, Linstrom's syphons on the tank and no tank valve. Axles are the "Coffin Process," the eccentric straps have Ajax bronze liners, as shown in our May issue (page 155), the smoke box is short, the guides are of the four-bar type and the design makes use of many of the details of the Class A engines illustrated on page 4 of our issue of January, 1896. The weight on drivers of the fast mail engine differed but 200 lbs. from the calculated weight, and the total weight was only 1,200 lbs. from that given by the designer.

Fast Mail Locomotives.
C. & N. W. Ry.

General Dimensions.

Fuel	Bituminous coal.
Weight in working order	133,800 lbs.
Weight on drivers	85,700 lbs.
Wheel base, driving	8 ft. 6 in.
Wheel base, rigid	8 ft. 6 in.
Wheel base, total	24 ft. 8 in.

Cylinders.

Diam. of cylinders	19 in.
Stroke of piston	26 in.
Horizontal thickness of piston	5½ in.
Diam. of piston rod	3¼ in.
Kind of piston packing	Dunbar.
Size of steam ports	18 in. x1⅝ in.
Size of exhaust	18 in. x3 in.
Size of bridges	1⅝ in.

Valves.

Greatest travel of slide valves	6 in.
Outside lap of slide valves	1¼ in.
Inside lap of slide valves	Clearance ⅛ in.
Lead of valves	3/32" lead at 6" cut off.

Wheels, Etc.

Diam. of driving wheels outside of tire	80 in.
Material of driving wheel centers	Cast steel.
Driving box material	Cast steel.
Diam. and length of driving journals	9 in. dia. x11½ in.
Diam. and length of Main crank pin journals	6 in. dia. x6 in.
Diam. and length of side rod crank pin journals	4½ in. dia. x4 in.
Engine truck journals	6 in. dia. x12 in.
Diam. of engine truck wheels	36 in.
Kind of engine truck wheels	Krupp steel tired.

Boiler.

Style	Extended wagon top.
Outside diam. of first ring	62 in.
Working pressure	190 lbs.
Material of barrel and outside of fire box	Carbon steel.
Thickness of plates in barrel and outside of fire box	11/32 in.; 3/8 in. 5/16 in., 7/16 in., 1/2 in.
Fire box, length	108⅛ in.
Fire box, width	40⅝ in.
Fire box, depth	front 78¼ in. back 64½ in.
Fire box material	Carbon steel.
Fire box plates, thickness, sides 5/16 in., back 5/16 in., crown 3/8 in.	tube sheet ½ in.
Fire box, water space	front 4½ in., sides 4 in., back 4 in.
Fire box, crown staying	Radial stays, 1 in. diam.
Fire box, stay bolts	⅞ in. and 1 in. diam.
Tubes, material	Charcoal iron.
Tubes, number of	320
Tubes, diam.	2 in.
Tubes, length over tube sheets	13 ft. 0 in.
Fire brick, supported on 5/8 in. Allison special iron tubes	18 in. thick.
Heating surface, tubes	2164.14 sq. ft.
Heating surface, water tubes	15.57 sq. ft.
Heating surface, fire box	173.68 sq. ft.
Heating surface, total	2353.39 sq. ft.
Grate surface	30.33 sq. ft.
Exhaust nozzles	4¾ in., 5 in., 5¼ in.
Smoke stack, inside diameter	16½ in. at top, 14 in. near bottom.
Smoke stack, top above rail	15 ft. 2 in.
Boiler supplied by	two injectors, Monitor "1888."

Tender.

Weight, empty	46,800 lbs.
Wheels, number of	8
Wheels, diam.	36 in.
Journals, diam. and length	5 in. dia. x9 in.
Wheel base	16 ft. 10 in.
Tender frame	10 in. channels.
Water capacity	5200 U. S. gallons.
Coal capacity	8 tons.
Total wheel base of engine and tender	51 ft. 11¾ in.

[We shall print a list of the dimensions of the heavy passenger locomotives in our next issue.—Editor.]

THE STEEL CAR SITUATION.

By R. P. C. Sanderson

Master Mechanic Norfolk & Western Railway.

A review of the difficulties experienced, now and in the past, in promoting the introduction of the Master Car Builders' standards, shows plainly that many of them were adopted too late and after individual companies, for lack of good standards to use, had put thousands of such parts in use, of an entirely different design, which they could not afterward afford to change.

Some of the old standards, which were abolished a few years ago, concerned old, light equipment, and were not suitable for modern heavier cars, and some of them were practically out of date by the time they were adopted. Those who have had most to do with the design of railroad equipment know that very few cases occur where an original design, no matter how carefully considered, is found to be just right, and the ruthless test of service soon shows that it must be strengthened here and changed there. A standard, if adopted in advance of experience with it, would, in all probability, be found to be defective and require changing in its details. There is a standing committee for the revision of the M. C. B. standards to correct errors, strengthen weak spots and practically keep the standards up to date; thus controlling the modification of the standards by the association instead of leaving each individual to alter them according to his own judgment.

The writer was one of those who, recognizing the above mentioned difficulties, and believing that the steel car frame was near at hand 4 or 5 years ago, hoped that by making a sufficient effort along proper lines, the M. C. B. Association could at least adopt some general standards, which would prevent the worst evils and annoyances that were bound to follow if each railroad company began to build steel car frames according to individual ideas. This would be of little consequence for special cars such as coal and ore hopper cars, as they must be designed to meet special conditions and seldom go off their own roads, or at least make the majority of their mileage on the home roads. It would be, however, of the greatest importance to have at least standard sections and lengths for the principal members of the steel frames for regular interchange cars, box, stock, long gondolas and flats. It would also enable the manufacturers to carry stock ahead, feeling certain of a sale for the same, and would avoid delays with the loss of service of valuable cars for weeks, while awaiting the manufacture and forwarding of some special shape.

After some preliminaries the Master Car Builders' Association appointed a committee to work on the subject, who, after a thorough canvass of all the roads and considering the recommendations of the traffic associations, etc., recommended certain leading dimensions for interchange cars, these now appearing in the published recommended practice to-day, and it is believed are about right for general conditions. It would be interesting to know if those who are building new interchange cars are paying the least attention to these recommendations, and if not, why not?

When the first committee on steel car framing reported to the convention it was evident that that body was not ready to commit itself to any one design, not having given the matter sufficient consideration, and the patent steel car interests were using what influence they properly could to prevent the adoption of a standard which would interfere with their market, and they were right from their standpoint. To dispose of the matter and to make some progress a fresh committee was appointed to submit individual designs, to meet the requirements. At the next convention those designs were submitted, and again the same condition of affairs was brought about, the convention was not prepared to act, had not prepared itself and could not decide such a matter off hand. Something had to be done, so a third committee was appointed to review these designs and make a final recommendation to the following convention. This committee recommended that the whole matter be dropped for four years, as they felt they lacked experience and knowledge to enable them to decide on any standard form of construction. They were perhaps right and felt that to get up standards for steel car framing might be too utopian an idea, and also perhaps too much of an interference with manufacturers of patented designs of cars. This was the death blow struck in the struggle to get any M. C. B. standards for steel car framing in advance, or at least contemporary with the demand. The results are already plain, and just what was anticipated. Some of the great railroads have placed orders for large numbers of steel framed cars of special designs, manufactured mostly of pressed steel shapes, and this even before a standard axle has been adopted for 100-000-pound cars, so that should the association adopt such a standard it will result in trouble. Some of the cars built may have such an axle under them, the others will not, and the roads concerned are not going to be too anxious to change them to conform to a subsequently adopted standard.

There are many different designs of 100,000-pounds cars and of details for these cars already in use, and more will follow. Each road will alter and improve their details and design, patented or otherwise, along their own lines, as experience directs, so that instead of the practice converging to uniformity it is diverging, and every year that passes makes the matter worse.

As regards the various designs and their merits, one could indulge in a good deal of speculation concerning the good and bad points of each, but as every design is patented, criticism might be considered as "offensive partisanship." Had the association seen its way to adopt one of the designs submitted there would at least have been a design left open for independent use, which is not now available. Any one can substitute steel eye beams or channels for wooden sills and call it a steel framed car, but that is not good design nor is it taking full advantage of the opportunities. Steel requires entirely different treatment in design from wood.

There are the Harvey patents covering steel sills with truss rods and bolsters of his design with multitudinous combinations of this and that, the Pennock patents covering the peculiar features of his designs, the Schoen and Fox patents, which are numerous, covering the use of pressed steel in almost every conceivable combination, and since the association did not want it and finally declined the design with thanks, the center girder design prepared by the writer and Mr. C. C. Wentworth, which was first freely offered to the association, has also been covered by patents for protection.

As the matter stands it would be exceedingly difficult to design a steel car frame now that would be worth the name which would not infringe somebody's patents. So that the steel car frame business is practically in the hands of the manufacturers. Naturally enough, the writer believes that the center girder and rib, spine and rib, keel and rib, design of frame originally suggested by him and illustrated in the M. C. B. proceedings, is the only correct principle for the design of flat bottomed cars, box, stock, flats, long gondolas, etc., especially for cars of large capacity, and others who have given the question deep thought, and who are not interested either way, directly or indirectly, have strongly approved the design. A number of such cars have been built to put the design to the proof and have shown up well so far. This is so because:

(1.) The greatest strength is in the center line of the car, furnishing ample material, disposed in the most effective way, to withstand any service shocks, and probably any reasonable collision shocks without injury.

(2.) There is no banjo work in the way of truss rods to require tightening; to be in unequal tension, to be bent or damaged in accidents.

(3.) There is nothing about the design but ordinary rolled merchantable shapes, purchasable in the open market, with the simplest kind of shop work.

(4.) The frames may be ordered in lots from any good bridge shop and shipped by the carload to the car shops or railroad shops to be unloaded off the cars by a hoist, dropped directly on to the trucks and run into the car shops to have the superstructure, brakes, etc., added.

(5.) There is the greatest possible strength for the minimum of weight (less weight than a weaker timber frame).

When we come to ore and coal hoppers, however, the problem is entirely different. There the sides must, perforce, be very strong to retain the load and to resist the lateral pressure of the load. The material, whether wood or steel, has to be there for this purpose anyway, and we should take advantage of the vertical strength lying dormant in this unavoidable material to help carry the load. We should add a little to the strength of the bolsters to transmit the load back to the center plate, making the center sills heavy enough to take the pulling strains and service shocks only; they need not carry any of the load beyond that small portion which lies directly over them where they pass through the hopper.

The question of whether in cars of this class the hopper bottoms and sides should be made of steel or of wood is one upon which there is a great difference of opinion, which it will take a good many years of experience to finally settle one way or the other. At the rate that cars of 100,000-pound capacity are now being put into service with steel hoppers and wooden hoppers, five or six years will give us a pretty positive answer to this question, if the railroads using them will keep record of the performance of these cars, the cost of repairs and the kind of repairs, which they are pretty sure to do.

[Before this article was written Mr. Sanderson expected to describe the steel frame cars to which he refers, but the drawings were not completed in time. They were received later, and an illustrated description of the design is given on page 187 in this issue.—Editor.]

MILEAGE OF CHILLED CAST WHEELS IN FREIGHT AND PASSENGER SERVICE.

For several years we have been favored by Mr. J. N. Barr, Superintendent of Motive Power of the Chicago, Milwaukee & St. Paul Railway, with the mileage records of chilled cast iron wheels in passenger and freight service. He has sent us the records for 1898 to bring these up to date.

In presenting this record in previous years we have directed attention to the fact that the average mileage is obtained by dividing the total car mileage during the year by the number of wheels taken out. If 10,000 wheels are in service, and 1,000 are removed each year, the average length of service would be 10 years, and the average mileage would be ten times the yearly mileage of the cars. As previously explained, this does not give accurate figures for any particular year, but it does give a correct method of comparison when a number of years are covered, and the statement shows the average mileage of all wheels taken out for all causes.

CHICAGO, MILWAUKEE & ST. PAUL RAILWAY COMPANY, OFFICE SUPERINTENDENT MOTIVE POWER.

STATEMENT SHOWING SERVICE OF FREIGHT CAR WHEELS.

Year.	Number of freight wheels made or bought.	Freight car mileage	Number of freight cars.	Number of freight wheels in service.	Average mileage.	Average life of wheels. Year.	Month.	Days.
1885	22,395	215,459,302	19,402	155,216	76,968	6	11	15
1886	19,459	236,140,449	21,385	171,080	97,080	8	9	15
1887	24,721	250,774,965	21,678	173,424	81.152	7	0	1
1888.....	24,162	261,400,022	22 544	180,352	86,544	7	5	17
1889.....	26,015	250,990,286	22,776	182,208	77.184	7	0	1
1890.....	15,823	263,983,845	23,864	190.912	183,468	12	0	24
1891	12.810	305,482,841	25,674	205,392	190,776	16	0	12
1892	17,340	334,943 674	26,308	210,372	154.528	12	1	18
1893.....	17,332	312,503 242	27,963	223,612	144,240	12	10	24
1894.....	11,647	276,390,355	27,800	222,400	189.784	19	1	4
1895.....	14,219	289,316,350	27,687	221 408	162 776	15	6	26
1896*.....	19,569	315,810,431	27,645	221,072	129,104	11	3	22
1897*....	14,634	292.285,995	27,517	220,048	159,784	15	0	13
1898*.....	19,420	344,752,791	30,120	240,876	142,020	12	4	25

* These records are on fiscal year basis.

STATEMENT SHOWING C., M. & ST. P. RY. CAST WHEELS IN PASSENGER SERVICE.

All Wheels Scrapped Except for Sliding.

Year.	Passenger. No of wheels.	Passenger. Average mileage.	Bag. mail and express. No. of wheels.	Bag. mail and express. Average mileage.	Parlor and sleeper. No. of wheels.	Parlor and sleeper. Average mileage.	Total. No. of wheels.	Total. Average mileage.
1885	840	46.068	811	45,641	65	42,708	1,676	45 731
1886.....	513	67,040	500	73,380	45	77.192	1,058	70,468
1887.....	391	80.243	470	88,379	28	96,366	889	85,053
1888.....	378	92,240	449	107,376	9	99.879	836	100,455
1889.....	455	100,864	477	112,031	9	141,706	941	106,916
1890.....	529	97.808	535	106.161	17	96,291	1,081	101.919
1891.....	892	96,919	885	109.673	51	102,605	1.828	103,252
* 1892.....	1,118	101,386	1,015	111,883	61	95 502	2,194	105.852
* 1893.....	1,098	99,139	1,092	110,858	29	122,998	2,219	105,218
* 1894.....	1,303	107,688	1,046	112,979	38	107,265	2,387	109,999
* 1895	1,144	106,949	1,002	114,542	48	93,715	2,194	110,127
* 1896	1,293	107,793	1,211	115,633	126	103,875	2,633	111,215
* 1897.....	1,518	101,248	1,439	110,136	182	99,994	4,139	105 250
* 1898.....	1,275	103,188	1,275	104,921	141	92.215	2,691	103,434

All Wheels Scrapped on Account of Sliding.

1885.....	640	23 936	513	18,218	96	12,514	1,279	20,654
1886.....	363	31 525	220	33.108	24	23 998	607	31,001
1887.....	337	44,467	324	48,352	12	68,080	673	46.759
1888.....	535	48,866	434	49,940	17	32,908	986	49.064
1889.....	591	51 596	346	60,705	4	49 874	941	54 936
1890.....	354	47 277	250	51,110	2	5 940	606	48 722
1891.....	466	49 142	271	48 573	44	38,770	781	48.360
* 1892.....	586	48 197	397	45.492	39	39 6 :8	1,002	46.831
* 1893	715	38,307	419	42,302	25	44.536	1,159	39.886
* 1894	653	43 353	366	42,842	44	29 679	1,063	42,610
* 1895.....	613	37,818	218	52 967	78	29,882	909	40,771
* 1896.....	382	38,280	219	43.188	125	32.392	726	38,747
* 1897.....	730	34,660	274	33,876	247	22,077	1.251	42,004
* 1898.....	736	36,724	292	45,096	182	30,529	1 210	37,819

All Wheels Scrapped.

1885.....	1,440	36 232	1.354	36,642	161	23.329	2 955	34.385
1886.....	876	52.322	720	61,047	69	58,618	1,665	56 359
1887.....	728	63,684	794	72,406	40	87,881	1,562	68,554
1888.....	913	66,827	883	79,146	26	56,091	1,822	72,645
1889.....	1,046	73 027	823	90,459	13	113,450	1,882	80,798
1890..... ..	883	77,437	785	88,629	19	86,780	1,687	82,750
1891.....	1,358	80 524	1,156	95,515	95	73,029	2 609	86,887
* 1892.....	1,704	82 508	1,412	92,133	100	73,727	3,276	86,479
* 1893.....	1,813	75,151	1,511	92,460	54	87.044	3,378	83,056
* 1894.....	1,956	86,207	1,412	94,799	82	65,634	3,450	89,235
* 1895.....	1,757	89,036	1,320	102,866	126	54,199	3,103	93,219
* 1896.....	1,675	91,840	1,430	104.538	251	68,674	3,356	95,568
* 1897.....	2,248	79,181	1,713	97,9:1	429	55,134	4,390	84,143
* 1898.....	2,011	78,863	1,567	933,773	323	57,457	3,901	83,080

* Fiscal year basis.

The average mileage for the past nine years is much larger than before that period, and there is a variation between 133,000 and 190,000 miles in the nine years. It has already been pointed out that between the years 1889 and 1890 there was a sudden jump in the figures, which is probably due to the introduction of the contracting chill, which has been used for a large proportion of wheels cast since that time.

The new Burlington fast passenger locomotives illustrated on page 141 of our May issue, we are informed by Mr. Delano, are not to be used in the fast mail service at present, but will go into the regular passenger and mail train pool between Chicago and Burlington, 206 miles, and they may be used for the fast mail service later. They will haul the Denver limited, leaving Chicago at 1.30 P. M., the time card schedule of which is 3 hours 55 minutes from Chicago to Burlington, with 11 stops. From Chicago to Aurora the train has 9 cars weighing 371 tons, and from Aurora to Burlington 34 tons less with 8 cars. Mr. Delano writes that in warming up one of these engines recently the run from Mendota to Aurora, 46 miles, was made in 48 minutes, with a 10-car train, weighing a little more than 400 tons, which is a remarkable performance for a new engine.

(Established 1832)

AMERICAN ENGINEER AND RAILROAD JOURNAL.

PUBLISHED MONTHLY

BY

R. M. VAN ARSDALE,

MORSE BUILDING.....NEW YORK

G. M. BASFORD, Editor.

JUNE, 1899.

Subscription.—$2.00 *a year for the United States and Canada; $2.50 a year to Foreign Countries embraced in the Universal Postal Union.*

Remit by Express Money Order, Draft or Post-Office Order.

Subscriptions for this paper will be received and copies kept for sale by the Post Office News Co., 217 Dearborn Street, Chicago, Ill.

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THE DESIGN OF CAST IRON CAR WHEELS.

With the exception of improving the metal and increasing the weight of cast iron car wheels, little or nothing has been done to improve their strength and ability to withstand the increased stresses brought upon them by the greater severity of shocks in present service, which are due to the use of cars of large capacity. An analysis of the stresses produced in cast iron wheels by the loading, brake shoe pressures and other forces is presented in this issue by Mr. Grafstrom, whose painstaking work in this direction suggests the importance of the examination of present practice in the design of wheels of this type with a view of improving them structurally. The motive power officers of roads operating over long mountain grades are exceedingly anxious about their wheels, owing to the combined effect of heavy loads and the heating of the brake shoes.

Notwithstanding the importance of this subject, very little has been done experimentally to show the effects of varying the design of wheels and the admirable report made to the Master Car Builders' Association in 1892 by Mr. H. J. Small of the Southern Pacific Company is very valuable in this connection, and his conclusions are more important now than when they were drawn, notwithstanding the improvements which have been secured by the introduction of the thermal test.

In the experiments referred to it was found that the cracking of the wheels was mainly due to the difference of temperature in the rim and hub. The cracks almost invariably started in the outer plate and within the junction of the inner plate. With the usual pattern of wheels the rim is expanded by the brakes and a force acts outwardly about in the plane of the middle of the single plate. On the outside of this plate there is not always enough metal to resist the expansion of the rim, while on the other side the rim is bound to the hub by the brackets, and these being exposed on three sides to the air, which is in rapid circulation, are more favorably situated for remaining at a lower temperature than corresponding parts of the single plate. The cool ribs seem to resist the expansion of the rim of the wheel on one side of the single plate and, there being no resistance on the other, the effect appeared to be to prevent the diameter of the tread of the wheel next to the flange from increasing as much as the outside of the tread. A contributing cause of cracking at the junction of the plates was the inferiority of the metal at that point on account of the porosity that resulted from the method of pouring.

Mr. Small suggested the use of less metal in the ribs, either by decreasing their number or their size, the use of the S bracket, which requires less force for extension of its length than the curved brackets and the extension of the single plate nearer the outside rim of the wheel. The results of tests showed conclusively that the design of wheels embodying these suggestions improved the ability to resist the effects of heating and when a large number of these were placed in service further support was obtained.

Mr. Grafstrom's analysis may be applied to the design of wheels and if combined with as careful experiments as those referred to it is probable that even better results may be had. The cast iron wheel has contributed in a very important way to the development of American railroads, and it seems wise that it should be improved to keep up with the increased severity of the demands imposed upon it.

A RESEARCH LABORATORY.

By M. N. Forney.

When the trees put on their summer attire and strawberries come to market, many of the readers of the American Engineer begin to think of the annual conventions of the two associations which assemble in June. This year they will meet at Fortress Munroe, where they have been held several times before.

Referring now to the Master Mechanics' Association, the list of subjects to be reported on are as follows: 1. A Research Laboratory Under the Control of the Association. 2. Boilers and Water Impurities. 3. Cast Iron and Steel-tired Wheels. 4. The Ton-mile Basis for Motive Power Statistics. 5. Staybolts. 6. Flanged Tires for Locomotives. 7. Best Form for Fireboxes, and 8, Nickel Steel.

The first subject is an old one, which was up for consideration twenty-six years ago. Inasmuch as there are now only a very few members who know what was done then, before the merits of the measure will be discussed, a brief résumé will be given of the action—or rather non-action—at that time.

In 1873, at the meeting held in Baltimore, a report was made on Water Purification, by a committee of which Mr. H. A. Towne was chairman. The report indicated that he had incurred some expense in preparing it, and Mr. Glass therefore moved "that Mr. Towne be reimbursed for the amount expended by him." No action apparently was taken on this mo-

tion. Afterwards another was made by Mr. Howard Fry, to the effect "that the committee be requested to appoint one of its members to receive from any member of the Association such specimens of water as he may require to be analyzed, to have it tested by some competent chemist, and return the anaylsis to the Master Mechanic sending it, the test to be made at the expense of the company with which the Master Mechanic is connected."

In discussing this motion Mr. Wilson Eddy observed, satirically, that "it would doubtless be a great satisfaction to our men in Boston to know that they can send water for analysis out to Omaha, or Chicago, or somewhere else. Possibly, though, they might think there were chemists nearer home who could do the job just as well. I think the railroads should select their own chemist to analyze their water, at their own expense."

Notwithstanding Mr. Eddy's opposition, the motion was carried, but afterwards Mr. Setchel moved "that no expense shall be incurred by committees except by the unanimous consent of the General Supervisory Committee, given in writing to the chairman of said committee, stating the amount to be expended." This also was carried. Later in the session, in discussing the need of a dynamometer, Mr. W. A. Robinson observed that "the time has come for the establishment of something of the kind to enable the members of the Association to make such practical experiments as they may deem necessary," and he then moved "that a Committee on Mechanical Laboratory be appointed by the President, and that this question be referred to that Committee." This motion also was carried, and later Mr. Robinson, Mr. Reuben Wells and Mr. J. M. Boone were appointed such a committee, which was afterwards increased by the addition of N. E. Chapman and H. M. Britton. At the meeting of the Association, in Chicago the following year—1874—this committee made an elaborate report. Accompanying the report was a communication from Professor Thurston, who was then connected with the Stevens Institute of Technology, in Hoboken, and one from the trustees of the Institute, in which it was proposed to establish a "Laboratory for Technical Research" in connection with that institution. It was proposed by the trustees to give the required space for such a laboratory, either within the present buildings or upon their grounds, and grant the use of all facilities for investigation possessed by the Institute. They also agreed to assist in the organization of the scheme, and, under proper restrictions, to assume the responsibility of its government and safekeeping. The committee very modestly estimated the cost of the apparatus at about five thousand dollars, "more or less," and apparently foundered on the rock of how even this modest sum could be raised.

A second paper by Professor Thurston on "The Necessity for a Laboratory for Technical Research" was appended to the committee's report. It contained among others the following wholesome observations: "The formation of a plan of operations requires a very definite knowledge of the character of the problem and an accurate understanding of the relative importance and probable bearing of the anticipated results." He also said—what seems hardly to have been realized by the advocates of the measure—that "The investigator must first know definitely what phenomenon it is proposed to investigate."

The trustees of the Stevens Institute reiterated their offer, and agreed to appropriate to the use of such a laboratory a strip of land adjacent to the Institute, having a value of about $20,000, on which the requisite buildings could be erected. Liberal offers of aid in organizing and conducting it were also made. The report with its appended communication excited a long and animated discussion. Mr. Eddy wanted to know how the money was to be raised, and said that "Many of us are liable to belong to the association only a year or two," and that the membership was constantly changing. Continuing, he added: "If we ask our respective companies to contribute to us, so that we can control apparatus for these various purposes, will not they tell us that they had rather we would use our education to determine a more economical way of hauling freight, or something of that kind, rather than to be spending our time and their money for something in the direction proposed."

The strongest argument was made by Mr. Coleman Sellers, who coincided with Mr. Eddy in thinking that "the society is a very evanescent one; it is one dependent upon people holding certain positions, and they remain active members so long as it is to their advantage to be mechanical engineers connected with railroads, but if they should leave that position, in all probability their interest would be diverted from it. The association exists totally different from such a society, as we may say, the Franklin Institute, which is a society having a location and building, and its members, of course, die off, and others are added to it, but its vocation remains fixed. Such an institution can very well collect about itself a laboratory and apply it to the use of its members. But there seems to me a great many difficulties in the way of attaching a laboratory to an institution that is merely formed with men who come together because they are in a particular trade, and that is the position held by this Master Mechanics' Association. . . . Properly fitted up laboratories exist all over the country, and to them these matters can be referred."

Mr. Jackman expressed the opinion that "unless we should do something more than superhuman, we could not accomplish very much more than what is accomplished from year to year by those who do examine these subjects, and publish the results of their examinations, so that the whole world can have the benefit of these results. . . . Inasmuch as there are many chemists employed all the time, and as they are publishing from time to time the results of the experiments they make, is it possible, or if possible, is it probable that we should get any better results than what we get through these chemists, who are all the time engaged in this business, and give us their tabulated statements, and who give us the results of all the experiments they make?"

Repeating, Mr. Sellers added, "The very fleeting nature of our association, the constant change of membership, and their wide separation, involves difficulties that it is very hard to overcome."

When the discussion was ended a motion was made to refer the subject back to the committee to report at the next annual meeting, which was carried. At the succeeding meeting, held in 1875, in New York, the committee reported that "After a careful reconsideration, they deem it the wisest course at the present time to recommend that this subject be postponed, indefinitely or until surer prospects exist of its being successfully carried out," and the project was then dead, to be resuscitated twenty-four years afterwards. The scheme then received careful and candid consideration, and most of those who advocated it, and the writer was one of them, were then convinced that it was impracticable. If it was so at that time, is it probable that it is less so now?

The weakness of the enterprise was pointed out by Mr. Sellers, and was then, and it is thought still, is due to the "fleeting and evanescent" tenure of membership in the Association, and the consequent lack of responsibility of those who might be called upon to take charge of its direction. Persons competent to successfully conduct experimental research are rare individuals, and to secure the services of such persons would require the most intelligent and disinterested discrimination and discernment. If a director of research was to be selected, what probability is there that the best or even a good man would be chosen? The world produces very few men competent to make original experimental researches, and the ability to recognize such capacity is almost equally rare. Now there are three ways by which a person could be selected for such a position by the Association. One would be to have him elected by the members—that is, by the counting of noses; or by appointment either by the executive or some special com-

mittee. In any one of these methods it is very far from probable that the persons whose votes would select the director of research would be entirely competent judges of the qualifications which such a person should have. The chances are that some one would be chosen for the position because he is a good fellow, or is a master mechanic out of a job, or some one who is the exponent of some interest, which could be served by the researches to be made.

The original scheme proposed by Prof. Thurston contemplated that "the personnel of the laboratory should consist of a director, with able assistants, good mechanics to take charge of tools and some unskilled labor." The employment of such a corps and their work would have involved the expenditure of a good deal of money, and this necessarily would have incurred a grave responsibility on the part of the Association. This was clearly recognized when the measure was fully and impartially considered twenty-five years ago, and as a consequence the association then declined to incur the responsibility, and the whole matter was "indefinitely postponed," or, as the committee said in their final report, "until surer prospects exist of its being successfully carried out." Since then the scheme slumbered until the meeting of 1897, when a motion was made and carried that a committee should be appointed to report at the next annual meeting on a "Research Laboratory Under Control of American Railway Master Mechanics' Association." Notwithstanding this action, this subject was not included in the list of those "for the 1898 convention" published in the first pages of the 1897 report. It is, however, the first in the list of subjects for the 1899 meeting, which is printed at the beginning of the 1898 report. Why this hiatus occurred in the appointment of the committee is not apparent. At any rate, the report to be read at the meeting this year will probably recommend substantially what the old committee proposed in 1874, with this important difference, that the estimated cost of the project has increased from $5,000 to $40,000 or $50,000, and the annual expense of operating or conducting it would be about $25,000 per year. It is also proposed that there shall be a manager in charge who is to be the Director of Research, assisted by a chief physicist or engineer in charge of physical tests; a chief chemist in charge of chemical tests. Under these men should be an ample corps of able and capable assistants. It is to be feared that with such a corps the annual expenses suggested would be quickly melted away in salaries.

That the end contemplated is a very desirable one is of course conceded. If an organization could be formed to which the doubtful and perplexing questions and problems which arise in the operation of railroads could be referred, and solutions obtained, it certainly would be a very desirable accomplishment. It is because it seems so very doubtful whether such an organization as is proposed will, or can, accomplish the ends aimed at that this article is written.

It may be suggested that the Executive Committee of the Association should have charge of the work of the Committee of Research. Now, the former committee, as Mr. Sellers said years ago, is an ephemeral one, its personnel and character is changed each year, and those composing it are not always selected because of their scientific sagacity, but much oftener merely as a compliment to members, because they are good fellows—which they generally are—and hold some official position on a railroad.

It would take too much time and space to describe fully here the qualifications which a competent Director of Research should have. To do anything which is worth doing he must be a real genius. When a person is recognized as a master of research, he at once becomes distinguished, and it is not then easy to hire the services of such a person. Let us suppose that it is decided to establish such a laboratory, and that in order to secure the services of the most competent person, the Executive Committee should determine to make their wants known and should undertake to formulate an advertisement for such a director. It can be imagined that it might read somewhat as follows:

WANTED.—A person to fill the position of Director of Research in the laboratory of the American Railway Master Mechanics' Association. He must have the following qualifications: A thorough technical education, and be able to write English clearly and lucidly (which most technically educated people cannot do), and know French and German. He must have had ample practical experience in a machine shop, and have a sufficient amount of inventive ability to be able, at all times, to "make a scientific use of his imagination" in the investigation of abstruse problems. Besides being an inventor, he must have the judicial and logical faculties sufficiently developed to be able to reason correctly and draw sound and unerring deductions from large numbers of phenomena and facts under his observation and ken. An important characteristic of his mind should be "dead-sureness," but allied with it there should be great alertness of body and mind for the apprehension of all kinds of phenomena, and their relation and effects. He must be able to take an enthusiastic interest in any subject presented to him for consideration and "research," and have unlimited pertinacity in the pursuit of the truth. Besides these qualifications, he must be amiable and civil, and capable of being all things to all men, and expect to be an unresisting target for the remarks of all sorts of ill-natured people who envy him his position. Added to these traits, he must be invulnerable to gifts and bribes and morally invisible, so that he cannot be "seen."

These are only some of the qualifications which a Director of Research should have. What chance is there of securing the services of such a man, at a salary of a few thousand dollars a year? Men of that kind may exist, but it is difficult to hire them; and yet the scheme is likely to fail unless it is in charge of some such person.

What has been said is not intended to imply a disbelief in the value of experimental research, but only to the method proposed for conducting it. As a matter of fact the British Institution of Mechanical Engineers has, for years past, conducted very successfully a series of the most valuable researches on various subjects. The method is simply this—some subject is selected on which research is desirable and more information is needed. A Research Committee is then appointed on that subject which is charged with the duty of conducting experimental investigation with reference thereto. Appropriations to meet expenses are made by the council, as seems to them judicious, and the committees employ such assistance as they require, of course with the sanction of the council. Committees of Research have been appointed on the following subjects: Boiler Seams and Riveting Friction; Marine Engine Trials; Steam Jackets; Alloys; and Gas Engines. Elaborate and very valuable reports have been made on each of these subjects.

Now, under this method of conducting research, the Institution is following the recommendation of Prof. Thurston, to wit: that "the investigator must first know definitely what phenomenon it is proposed to investigate." It would, it is believed, be much wiser and more practicable for the Master Mechanics' Association to act in accordance with this advice, and the practice of its sister organization, than to embark in an extravagant scheme which will lead to results which no one can foresee, and which would be liable to abuses which might bring discredit to the Association and its members.

The plan of the Institute of Mechanical Engineers is very simple, and could very easily be adopted and tried, and, if successful, could be extended, or if it failed would not incur any disastrous consequences. Such a plan was proposed in these pages a year ago, in an article on Competitive Tests of Locomotives. It was then suggested that a series of such tests should be made under the auspices and direction of a committee to be appointed for that purpose—the committee to formulate a plan for making the tests, and have charge of them, and

direct how they should be made. As such tests would occupy weeks or months, the committee would not be able to give the time required to personally conduct the series of experiments, and therefore it was suggested that they be authorized to employ some competent person or persons to take charge of them, of course under the direction of the committee. In the article referred to it was said further that "probably no competent person could be found to take charge of such work who would be willing to give the time to them which would be required without compensation, and therefore the mechanical expert and his assistants would have to be paid. This would imply that some money must be provided for such, and some few other expenses. The committee should therefore be authorized to raise and expend money for the purpose contemplated, with the usual powers and responsibilities of an auditing committee."

In his annual address last year, President Leeds made a similar recommendation, but it was not acted upon. It is urged here that such a measure would inaugurate the system of Experimental Research, so successfully adopted and practiced by the British Institution of Mechanical Engineers, that it could be done with very litle cost or risk of disaster, and, if successful, the method of investigation could be extended to other subjects; and, if the appliances of a mechanical or chemical laboratory were at any time needed, those already in existence could be resorted to; and there can be little doubt that the services and assistance of the persons in charge of them could in most cases be obtained. It may very well be that in the prosecution of such research that some special apparatus or appliances not to be found in any of the laboratories might be required. If that should be the case, it will then be early enough to consider the ways and means of getting what is needed.

The purpose of this article is to suggest that the committee making the report this year should accept Prof. Thurston's recommendation and formulate "definitely what phenomena it is proposed to investigate," and then have a special committee or committees of research appointed to investigate such phenomena, utilizing as far as possible existing laboratories, without contemplating any expenditure for experimental appliances, excepting for such kinds as are not now in existence and may be needed in the future.

A good subject to begin on would be "Competitive Tests of Locomotives," which was proposed last year.

So much room has been occupied in the discussion of the first subject for consideration at the Master Mechanics' convention that it is impracticable to consider the others, which will be reported on, and which it was at first intended to discuss in this article.

THE IMPORTANCE OF HIGH PRESSURE IN FUEL OIL BURNING.

Almost any morning outside of a certain large manufacturing plant in Philadelphia in a side street may be seen a heap of black refuse, which has just been dug out of some 50 or more oil furnaces used in the works in forging, etc. Ordinary passers-by would be shocked to learn that this represents a value in fuel, that has to be paid for, of over $60 at a very moderate estimate. This waste is the carbon which forms in such furnaces where the combustion is incomplete. The fires in this case are smoky and the furnaces are soaked with oil that is spattered about, indicating at once that something is wrong with the method of burning the oil, although we are told that this is most excellent in efficiency and "entirely satisfactory."

Contrasted with the above the writer inspected some oil furnaces in a little country place, where few persons would see them without a special pilgrimage. It was a small plant with only five or six furnaces. There was no smoke, no smell, and the fires were at a white heat, turning out evidently at least three times the amount of work per hour that could be done in the ones first described. On rubbing a white handkerchief around the glazed walls inside of one of the furnaces, which happened to be out of use and cool, no spot or soil could be found on the handkerchief, and this indicated complete combustion.

This system, like the first, is an evolution, but an evolution from a different root. It could not have come from the low pressure system, for all its principles of action are unique. It is rather the outgrowth of the experiments instituted in the 5 or 6 years just past, when compressed air was discovered to be possessed of the properties required for burning oil properly. These experiments have all in turn been rewarded by quick responses, and a new set of progressive data obtained from them, all in the direction of increased economy. It may prove instructive to consider the two best types of low pressure and high pressure systems of oil burning.

The low pressure system is operated by blower blast, and as the pressure is only about 6 to 8 ounces, only a very ineffectual breaking up of the oil can be attempted. Much of the fuel therefore is blown in to the furnace in solid drops, from which low heat efficiency is obtained. The finer portions of the spray, mixed with the air in the blast, may burn to produce a temperature as high as 2,500 degrees, but rarely so much as that, and this will mean about 4,000 heat units per pound of oil burned. The drops make carbonic acid. If this carbon could all be burned the temperature of the furnace would rise to over 4,000 degrees and the heat units would be 12,000 per pound; this is a difference of 8,000 units which are lost. These are sufficient to heat 16 pounds of iron to a welding heat, or in one furnace it would add 4,800 pounds to the output of work per day. This is a pretty large loss, unless the carbon can be sold at what it costs to make.

In the second case mentioned, which may be very properly regarded as the highest development of the compressed air systems, the first remarkable feature seemed to be the fine atomization, under an unusually high pressure, 25 to 30 pounds, the oil being delivered at just enough pressure to insure a solid supply at the burners, about 5 or 6 pounds. The jet formed was driven through the atmosphere from about 5 inches away from the funnel entrance to the furnace, inducing free air in a ratio due to the velocity of the jet, and it was lighted with a torch. The combustion seemed to be absolutely perfect from the start and the temperature picked up very rapidly to a high degree, constantly increasing until the furnace was white hot. As there was no smoke and no carbonic oxide formed, there must have been, according to the laws governing carbon combustion, a temperature making carbonic acid, and as there is 84 per cent. of carbon per pound of oil, this would mean 84 per cent. of the heat units belonging to the pound of carbon under perfect combustion, or 84 per cent. of 14,500, which means 12,180 heat units secured.

There is also 14 per cent. of hydrogen liberated from a pound of oil and it is fair to expect that by the extra induction of air at the right point this combustion could also be and probably was secured, which gives 14 per cent. of the 62,000 heat units belonging to the pound of hydrogen, or 8,680 units. Then 12,180 (the carbon heat units), plus 8,680 (the hydrogen units), gives a total of 20,860, or all there is to be gained from the fuel (the 2 per cent. of nitrogen in the pound of oil being valueless.

This seems to be an ideal method, and although probably no furnace can be constructed to utilize all this heat, yet there seems to be encouragement to expect that at last something like finished results are in sight from the use of oil fuel. A description of the high pressure method whereby 30 pounds of iron were heated (to a welding heat), per pound of oil, at the Juniata shops of the Pennsylvania Railroad at Altoona is given on page 42 of our issue of February, 1897. It is perfectly safe to say in this connection that very few people who are using oil know the ratio between the amount of iron heated and the quantity of oil burned. It is also safe to say that in ordinary low pressure systems not more than 15 pounds of iron are heated by each pound of oil.

The well-known firm of H K. Porter & Co. has been succeeded by H. K. Porter Co , with Mr. H. K. Porter President.

CAR WHEEL CONSTRUCTION.

By Edward Grafstrom.

A great deal of progress has been made in recent years in the manufacture of cast iron car wheels. With the growing demands of traffic the railroads have from time to time tightened up their specifications and insisted on more stringent tests. New requirements have generally been met by changes in the metallurgy of the wheel, while its mechanical construction and its form has practically remained the same as the original Washburn or Atwood pattern, which, since the drop test drove the single-plate wheel off the market, has become the generally accepted standard for ordinary service.

The introduction of contracting chills did not bring about any change in the form of the wheel, nor did the adoption of the thermal test. The broad tread wheel with its 6¼-inch rim has passed, and the Master Car Builders' Association has defined the contour of flange and tread. The length and bore of the hub are determined by the axle, and weight limits have been decided upon. Otherwise there is no general agreement between the railroads and the manufacturers as to the form of the wheel, yet there seems to exist a tacit understanding that the Washburn type with ring core and curved brackets is the proper wheel of the present day. The size and shape of the ring core may vary, as well as the number and curvature of the brackets, and the thickness of the plate, but the general design remains the same. With the introduction of 80,000 and 100,000 pounds cars stronger wheels were needed. The old patterns were re-enforced a little all around, the plate was made heavier and perhaps a couple of brackets were added.

The writer has recently been asked the question more than once where added metal will do the most good on a car wheel. The scrap yard gives the reply in a way, but to get the best distribution of the iron requires a careful analysis of the stresses which may be produced in the wheel, as far as they can be determined. Those caused by external forces are more or less known, but the internal strains are apocryphal; to what extent they are reduced in the annealing pit is uncertain, but it is here suggested that if, by suitable manipulations of the metal while cooling, these stresses can be brought under control as to magnitude and direction, they may, instead of being detrimental to the structure, add strength to it by counteracting the effects of the external forces, in the same manner as the initial strain produced by the camber assists the car-sill. For the present, however, these stresses must necessarily be left out of consideration.

The writer has never yet seen in print any calculations for a car wheel; the following analytical method is therefore offered, the value of which depends wholly upon the accuracy of the data available. For the purpose of representing extreme conditions surrounding a car wheel in actual service under a 30-ton car, a box car is assumed weighing 32,000 lbs. and carrying the maximum load of 66,000 pounds, making a total of 98,000 pounds. The car-body weighs 21,000 pounds, and its center of gravity, as recently determined by Mr. A. S. Vogt, of the Pennsylvania Railroad, lies 5 feet 6 inches above the top of the rail. The two trucks weigh together 11,000 pounds, and their center of gravity is, according to the same authority, located 19½ inches high. For the lading the corresponding distance is assumed to be 6 feet. The center of gravity of the total weight of 98,000 pounds will thus be found to be 5 feet 4.82 inches high, as shown in Fig. 1.

Going at a rate of 60 miles per hour around a curve of 1,000

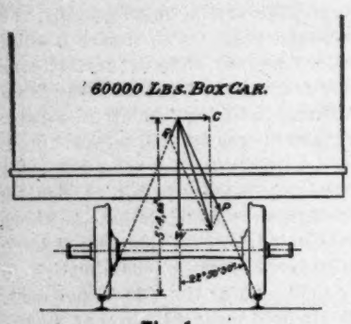


Fig. 1.

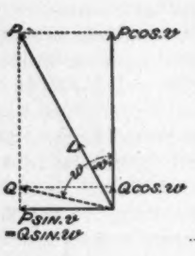


Fig. 4.

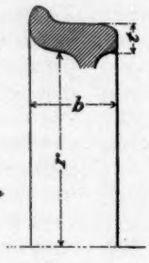


Fig. 5.

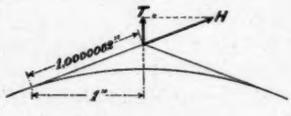


Fig. 6.

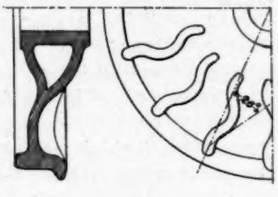


Fig. 7.

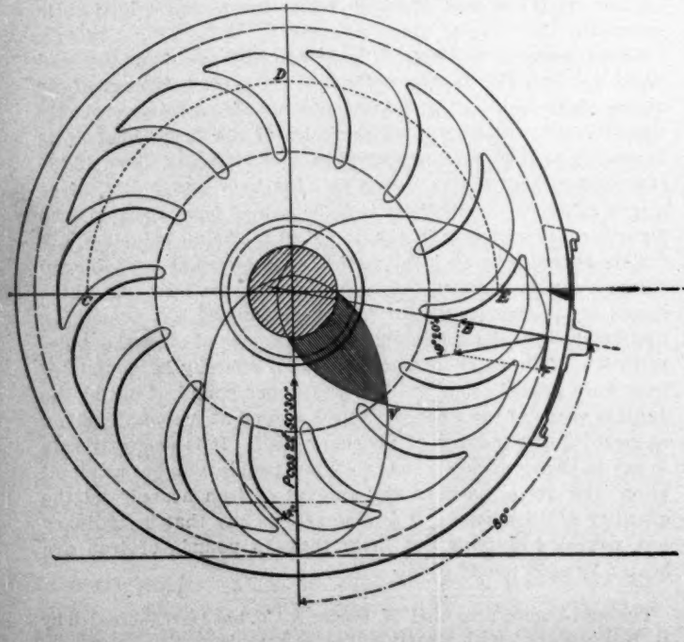
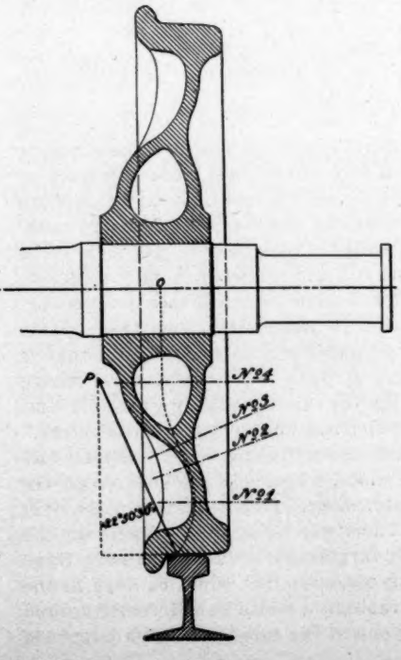


Fig. 2.



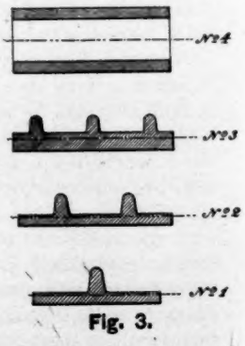


Fig. 3.

feet radius, the centrifugal force C of this car would be equal to 23,570 pounds, and the resultant of this force and the weight W would make a total of 100,800 pounds. If this is divided into two components, one through each rail-head, the pressure P on the outside rail would be 82,054 pounds, acting at an angle of 22 degrees, 50 minutes and 30 seconds with the vertical. The reaction on each wheel, also marked P in Fig. 2, is then equal to 20,514 pounds. The component P_1 acting on the inside rail, being only 6,463 pounds per wheel, will not be further considered. The force P is transmitted to the axle partly by the metal located between the rail and the wheel-fit, which is thus under compression, and partly through the rim to the diametrically opposite portion of the wheel, whereby a tension is caused in the plate between the rim and the hub, in the same manner as the spokes of a bicycle wheel are put in tension by the load at the center. The portion of the wheel under compression will be considered first. This portion does not include the whole lower half of the wheel, but only a section of it within a limited distance from the center line of strain, which connects the point of contact between wheel and rail with the center of the axle. The form of this section may be called a cycloidal sinoid, the abscis equation of the curve corresponding to the cycloid, while the ordinate equation is that of a sinoid, acording to Reuleaux. The outline of this form, as shown in Fig. 2, is described by Redtenbacher's formula:

$$\frac{x}{l} = \frac{1}{\pi}\left(\text{arc. sin } \frac{y}{h} - \frac{y}{h}\sqrt{1 - \frac{y^2}{h^2}}\right),$$

where it should be observed that the diameter of the wheel-fit must conform to y, in order to enclose all parts of the wheel under compression.

This cycloidal sinoid can now be considered as a column, secured at the upper end, and its strength is expressed by Gordon's modification of Hodgkinson's formula:

$$K = \frac{F \cdot S}{1 + 1.8\,a\,\frac{l^2 \cdot F}{J}},$$

in which F represents the area of cross section, J the moment of inertia, l the length, S the stress per square inch, and "a" a constant relating to flexure, which, according to Wm. Kent's new hand book, may be assumed as one sixty-four hundredth for cast iron.

In order to find the value of K four cross sections of the column have been selected at different places, as marked in the sectional view of Fig. 2. These cross sections are also shown in plan in Fig. 3. It will be noticed that the number of ribs falling within the cycloidal sinoid varies in the first three sections, and that section No. 4 is taken through the double plate. The type of wheel illustrated in Fig. 2 is the Pennsylvania Railroad 1895 model, which is the standard for 60,000 pound cars on a number of roads. The curvature of the brackets on this wheel is very marked for section No. 1, a tangent drawn through the section-plane making an angle of 42 deg. with the vertical. The bracket makes up 79 per cent. of the cross section area, and for this section K must therfore be substituted with $0.21K \times 0.79K$ sec. 42 deg.

It will also be observed that if another cycloidal sinoid is described for the brake shoe pressure, it will overlap the one for the rail-reaction, as indicated by the shaded surface in Fig. 2. The inclination of the section-lines also shows the resultant direction of the rail-reaction and the brake shoe pressure combined, and this resultant must be considered in connection with section No. 4. The brake shoe pressure marked B in Fig. 2, being 70 per cent. of the light weight, is equal to 8,675 pounds, and its direction is assumed to be 80 deg. from the vertical. The coefficient of friction at 60 miles per hour, is not more than 16 per cent. (report of M. C. B. committee on laboratory test), which makes the frictional resistance F = 1,372 pounds. This and the brake shoe pressure result in a force of 8,684 pounds, deflecting 9 deg., 20 min. from the latter, or 49 deg., 20 min. from a line drawn from the center of the wheel to the point V, where the two curves intersect in Fig. 2. The cosine of this force for 49 deg., 20 min. added to the cosine of K for a 40 deg. angle, will therefore take the place of K in Gordon's formula for section No. 4.

As the column is not loaded squarely, however, but at the angle of 22 deg., 50 min. and 30 sec., the formula must be further modified in order to cover the direct compression as well as the bending strain. If the load of the column is called L, K becomes equal to L cos. 22 deg., 50 min., 30 sec., which represents the compression, while L sin 22 deg., 50 min., 30 sec. stands for the bending component. Let v signify the angle, and the formula will appear thus:

For section No. 1:

$$L\left(0.21 + 0.79 \text{ sec. } 42 \text{ deg.}\right) = \frac{S \cdot F}{\text{cos. } v\left(1 + 1.8\,a\frac{l^2 F}{J}\right) + \frac{F}{Z} l \cdot \sin v}$$

For sections Nos. 2 and 3:

$$L = \frac{S \cdot F}{\text{cos. } v\left(1 + 1.8\,a\frac{l^2 F}{J}\right) + \frac{F}{Z} l \sin. v.}$$

For section No. 4:

$$\left(L. \text{ cos. } v. \text{ cos. } 40 \text{ deg } + 8684 \text{ cos. } 49 \text{ deg. } 20 \text{ min.}\right)\left(1 + 1.8\,a\,\frac{l^2 F}{J}\right)\frac{F \cdot l}{Z} L \text{ sin. } v = S. F.$$

It should be stated here, that as v signifies the angle which the direction of the force forms with the neutral plane at each section, and as at sections Nos. 2 and 3 the neutral plane recedes from the vertical, its angle of deflection must be deducted from v. For section No. 2: v—18 deg., 30 min., and for section No. 3: v—21 deg., 45 min., will therefore be used in place of v alone. The neutral plane is shown in the sectional view of Fig. 2 as a dotted curve extending from point O on the center line to the point of rail-contact.

The length l is measured perpendicularly to each section plane from the point of rail-contact. By inserting the values of F, J and Z for each separate section in the formula applying to the same, L can now be obtained for each of the sections. It will be found that its lowest value is for section No. 1, where L = 0.7842....S.

As before stated, S represents the stress per square inch, and, if J and Z have been calculated for the portion of the column located to the left of the neutral plane in Fig. 2, this stress denotes the compression wholly. According to tests made a few years ago by Prof. Thurston, the ultimate resistance to compression is as high as 127,300 pounds per square inch for Salisbury car wheel iron; but for the purpose of this calculation it is assumed to be 120,000 pounds. If the force is applied suddenly, 28 per cent. of this, or 33,600 pounds, is, acording to Nystrom, all that can be counted on. Allowing now a factor of safety of 4, the safe compression stress in the wheel would be 8,400 pounds per square inch. If this value is inserted for S in the above expression, L becomes equal to 6,588 pounds, or in other words, the cycloidal sinoid of the wheel can safely stand a suddenly applied force of that magnitude, acting at an angle of 22 deg., 50 min., 30 sec. from the vertical.

Another force Q (see Fig. 4), at a certain angle w, may be substituted for L, as long as the sum of the horizontal and vertical components of each is the same, or: L sin. v + L cos. v = Q sin. w + Q cos. w. If w is chosen sufficiently large to make Q sin. w coincide with the rail-reaction's horizontal component P sin. v, the latter can be substituted for the former in the above equation, from which then the value of Q cos. w = 131 pounds can be obtained. Subtract this from P cos. v or 18,615 pounds, and the remaining 18,484 pounds are the part of the rail-reaction which, being transmitted through the rim to the upper half of the wheel, causes a tension in the plate.

Before proceeding further the ability of the rim to transmit this stress will be ascertained. For this purpose the following empirical formula is used:

$$M = 6{,}600{,}000.\ b\ \frac{(t-c)^2}{r^2}$$

in which b, t and r represent the dimensions indicated in Fig.

5; c is a constant which Nystrom gives as ½ for cast iron; and M is the ultimate sustaining strength against collapse of one-half of the rim. By solving the equation M is found to be 544,500 pounds, which is more than sufficient for the purpose.

Returning, therefore, to the tension in the upper portion of the wheel, attention is now called to the generated forces which augment the stress caused by the rail-reaction. These are the centrifugal force of the wheel, and the strain caused by the expansion of the rim due to the heat produced by the brake shoe friction. Still another strain, induced by pressing the wheel on the axle, must also be considered. The latter strain, together with that due to the centrifugal force, tend to burst the wheel along its diameter, while the tension caused by the rail-reaction, the expansion by heat and the centrifugal force aim to produce rupture along the semi-circular section indicated by the dotted line C D E in Fig 2, where the plate is weakest.

The total centrifugal force of a 595 lbs. wheel at a speed of 60 miles per hour is equal to 80,474 pounds, if the square of the radius of gyration is 0.6 of the square of the radius of the wheel, as stated by Mr. R. A. Parke (N. Y. R. R. Club Proc., Nov., 1897). The bursting strain caused by pressing the wheel on the axle with a force of 30 tons, would, with a coefficient of friction of 0.2, come close to 300,000 pounds, according to recent experiments at the Alabama Polytechnic Institute by Prof. Wilmore, although earlier tests by Lucien Arbel put it considerably lower. The sum of this bursting strain and the centrifugal force, divided by π, represents a force of 121,220 pounds, which tends to burst the wheel in two along its diameter. The cross section area along this line is found, according to Guldin's law, to be 64.88 square inches for a 595 pounds wheel, if the specific gravity of the iron is 7¼. The strain per square inch of section is thus equal to 1,868 pounds.

Considering now the stresses which tend to rupture the plate of the wheel along the line CDE in Fig. 2, there is first the tension of 18,484 pounds due to the load. As before stated, this may be of the nature of a suddenly applied force, and the fiber strain caused by it is therefore from 1.6 to 2 times greater (Nystrom). Selecting the latter figure, we have then a tension of 36,968 pounds, which will cause a radial stress of four times as much or 147,872 pounds (Rankine). Next there is the centrifugal force of the metal lying outside the line CDE. Assuming this line to be at a distance of 13½ inches from the center, the centrifugal force of the mass will be 24,693 pounds. Finally we have the strain due to the sudden expansion of the rim on account of the generated heat. Assuming an extreme difference of 300 degrees Fahr. in the temperature of the rim and of the plate inside section CDE, a modulus of elasticity of 17,000,000 a co-efficient of expansion of 0.0000062, and inserting these values in the following equation:

$$H = A. \times 0.0000062. \times 300 \times 17{,}000{,}000,$$

in which A represents the cross section of the rim, we get H = 347,160 pounds. Referring now to Fig. 6, and considering the rim as a polygon composed of a number of surfaces one inch in length, each one of these would increase to 1.0000062 for each degree of temperature, causing a radial strain T in the plate The size of T is found thus:

$$H : T = 1.0000062 : \sqrt{1.0000062^2 - 1},$$

which gives T = 0,00346H for each inch of the circumference, or 25,500 pounds for the whole distance CDE.

Adding now the tension from the load. 147,872 pounds, the centrifugal force, 24,693 pounds, and the expansion strain, 25,500 pounds, we get 208,065 pounds as the total radial strain which tends to produce a rupture along the line CDE. The transverse area at this distance from the center is 44.26 square inches. The stress per square inch is therefore equal to 4,927 pounds, or about 2½ times as much as in the diametrical section calculated before. As the sectional area through the line CDE also cuts through the brackets, which at that plane incline 42 deg., the stress per square inch of metal in the brackets is proportional to the secant of the angle, thus bringing it up to 6,631 pounds. If the tensile strength of the iron is 22,000 pounds, we have therefore a factor of safety of only 3½ in the brackets. In order to get the same safety in tension as in compression, iron with a tensile strength of 26,524 pounds would be required. As that is more than can be expected from the common grades of car wheel iron, the brackets in the wheel are the weakest places and will fail oftener than other parts.

Fig. 7 shows the standard Union Pacific wheel. It will be noticed that the tangent to a bracket at the dangerous section forms here an angle of only 26 deg. with the radius, which makes the secant of the tension in the bracket considerably less than in the previous example, bringing it down to 5,474 pounds, a factor of safety of 4. This points to the advisability of avoiding unnecessary curvature of the brackets.

It is interesting to note that if the method used in the above calculations for finding the stress due to the expansion of the rim by heating be applied to the thermal test of the wheel, it will be found that a difference of 2,448 deg. Fahr., between the temperature of the rim and that of the adjoining plate, is necessary to produce the same stress of 4,927 lbs per square inch as that caused by the assumed conditions of service used in these calculations. The average melting point for gray iron is, according to Greenwood, 3182 deg. Fahr., which leaves a margin of 734 deg. for the temperature of the plate, and also for the difference in temperature between the molten metal and the rim itself.

On many roads it is the custom to allow wheels with cracked brackets to remain in service. The analysis shows that this is not a safe policy, unless temporary limits are established for the load and the speed. If the brakes are cut out on a car, one-eighth of the possible maximum tension is eliminated, and a proportionate number of cracked brackets will not therefore impair the safety of the wheel, unless they are bunched.

By observing the ratio in which the various stresses in the wheel affect its strength, the most economical wheel section for a 100,000 pounds car may be developed, and it will also be seen how a wheel might be proportioned for the special purpose of withstanding the thermal test, regardless of service conditions.

In conclusion it should be said, that nothing new is claimed in this method of analyzing the stresses in a wheel; it merely represents a consecutive application of well known formulas, as in designing other mechanical constructions.

INCREASED CAPACITY OF THE PRESSED STEEL CAR COMPANY'S WORKS.

A recent Associated Press dispatch from Chicago stated that a new steel car plant was to be established in or near that city. This probably originated from the fact that the Pressed Steel Car Company is to greatly enlarge its plant at Joilet, Ill. A large tract of land has been purchased at Joilet, and the old Fox plant, now the western plant of the Pressed Steel Car Company, is to be enlarged to at least three times its present capacity. Ground has already been broken at Joilet, and the work will be pushed with the vigor characteristic of the company. When completed, the enlarged plant will have a capacity of 30 pressed-steel cars a day, in addition to a proportionate increase in the capacity for trucks, bolsters and other pressed steel parts. The total investment will aggregate $500,000, and about 1,600 skilled workmen will be added to the force. With the Joilet plant complete, the capacity of the various works of the company will be in excess of 100 pressed-steel cars per day.

The rapid growth of the company is of considerable interest as well as importance. Within two years in Pittsburgh alone, the company has added 4,000 men to its force, and, as may be well understood, its pay roll has increased many thousands of dollars weekly. The new plant at McKee's Rocks, Pittsburgh, which alone will double the capacity of the Pittsburgh plants, called for an investment of over one million dollars. This plant will be in operation in July and will have a capacity of 40 pressed-steel cars, in addition to a large bolster and truck capacity. The Joilet plant was designed to take care of a share of the Western demand, which has increased so rapidly that the company has found extreme difficulty in caring for it. Since the first pressed-steel car was built, orders filled and unfilled aggregate about 15,000 cars.

PISTON VALVES FOR LOCOMOTIVES.

By F. M. Whyte,

Mechanical Engineer Chicago & Northwestern Railway.

It was early appreciated that the main valves of locomotives ought to be so constructed as to be operative with as little the strain as possible on the valve motion, and while it would appear that the use of what is termed the balancing on the usual form of the D slide valve has been more generally adopted in America, nevertheless, the importance of having a thoroughly balanced valve was understood in England and Europe years ago and efforts were made toward developing such valves. The familiar saying that "History repeats itself" is true when applied to locomotive engineering, and it is frequently remarked by old railroad men, that: "I saw something just like that when I was serving my apprenticeship," when their attention is called to something which is considered new at the present time. So it is with piston valves; it is extremely interesting to study the history of piston valves, and other designs made with the intention of decreasing valve friction, and to note that engineers of the present are learning some things about valves which were learned by the engineers of fifty or sixty years ago, and, while conceit may lead to the belief that much of the chaff has been discarded, it is quite possible that the engineers of a few years hence may find much that is now being overlooked. This, however, need not reflect on the present engineers, because development of other parts of the locomotive may make designs desirable which are not now suitable.

Among the early attempts to reduce the friction of valves may be mentioned that of Mr. John Hick; Hick placed rollers under the valves, the valves being carried on journals which were reduced below the diameter of the body of the rollers, the body of the rollers bearing directly on the valve seat. In 1859 Mr. Thos. Felton patented in England, and, in 1863, Capt. R. C. Bristol, of Chicago, patented in the United States, the idea of interposing the rollers immediately between the valve face and the valve seat, instead of carrying the valve on the journals of the rollers. Valves so supported were used in locomotive and marine service, and technical papers of that time report that valves so arranged gave much satisfaction, no difficulty being experienced in keeping the valves tight. There is nothing to indicate whether such reports emanated from disinterested or from interested persons, but the average railroad man of the present, guided by present demands and experience, would conclude that the reports were made by the former. Later, in 1868 and 1869, the Messrs. Sault, of New Haven, used rollers under valves, the rollers varying in diameter from 4 to 12 ins., depending on the size of the valves, and the use of such design in locomtive service was favorably reported.

The advantages which possibly may be obtained by the use of piston valves were early appreciated, and also the same difficulties which are experienced now with some designs of piston valves were met years ago and efforts were made to surmount them. These valves were called "equilibrium valves," and in some cases the very expressive adjective "perfect" immediately preceded the word "equilibrium." It is quite probable, however, that the fact, which is sometimes overlooked at present, the equilibrium of a piston valve may be very seriously affected by the use of packing rings, was understood, because means were provided for taking up wear; this might possibly indicate that the valve was made to fit closely in the casing, and this conclusion is strengthened by the direct statement made in literature of 1866 to 1869 to the effect that much difficulty was experienced by the unequal expansion of the piston valves and the cylinders in which they were placed. Apparently appreciating this difficulty, Mr. Thos. S. Davis, of Jersey City, designed in 1866, or immediately previous to that date, the valve and valve chest shown in Fig. 1; the valve casing, or cylinder in which the valve worked, was almost entirely surrounded with steam and the piston valve could be adjusted to compensate for the wear. The valve was made long so that the passages between the steam chest and cylinder would be as short as possible. This was one of the early designs of valves which were claimed to be "perfect equilibrium valves," and it is believed that in this valve there were embodied the essential features of a successful "perfect equilibrium piston valve;" certainly there were in the mind of the designer those features which are now considered requisites, and his efforts were directed toward fulfilling them, whether he succeeded or not. The valve had no packing rings, but could be adjusted to compensate for wear; the valve casing, or cage, was so arranged that most of its outside surface was subjected to the temperature of the steam at boiler pressure; and the valve was of such length as to give the shortest ports possible.

This may be an appropriate place to declare the belief that a piston valve which is to fulfill the strongest claim made for such valves, that they are perfectly balanced, can not have the ordinary, nor many of the extraordinary, spring packing rings for making the valve tight, and an effort will be made later in this article to show what effect such rings, in some of the present designs of piston valves, have on the balancing feature. If this declaration is well founded, and it is demonstrated that the valve must be solid and without rings, then it follows that the valve must be a neat fit in the valve cage and the cage must expand and contract with the valve; this will require that the cage be subjected to about the same temperature as the valve, and that provision be made to allow free expansion and contraction of the cage.

In Fig. 2 another "equilibrium valve" is shown, which closely

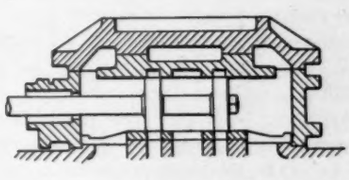


Fig. 1.

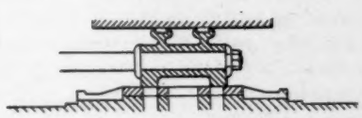


Fig. 2.

resembles an adaptation of the present common balancing plate and strips. The valve was designed by Mr. W. G. Beattie in 1867, or a short time previous, and was used on a locomotive on the London & Southwestern in 1868. Then, as now, experiments were made at as low cost as possible by first adapting new ideas to old equipment, and the illustration shows the valve as applied to a cylinder casting which was not designed specially for the valve. The valve was placed on a locomotive in January, 1868, and ran 16 months, making 61,147 miles, and showed so little wear that it was in condition to be placed directly in service again.

Apparently the adaptation of the piston valve to locomotives was neglected for a number of years, and the cause for such neglect is not apparent. It may have been considered that the benefits to be obtained were not commensurate with the inconvenience in changing designs of other parts, and of course this would be a more serious question at the present time. However, perhaps enough of history has been given to show that we are "threshing old straw," and it will be more to the point to show the results of the present threshing.

Until very recently it has been the general practice to use spring rings to make steam tight joints between the piston valve and its cage, or containing cylinder, and that such rings may fulfill the purpose for which they are intended it is necessary that they fit loosely between the metal forming the two sides of the grooves in which they are placed and the wearing action between the rubbing surfaces accompanied by the intended springing of the rings outward to compensate for the wear, soon leaves spaces back of the rings, even though the

rings, when new, may bottom in the groove. On account of the space required at the sides of the rings and behind them, it is very certain that the rings are pressed outward by the pressure of the steam in the steam chest, as well as by the springing action of the rings, and, indeed, this is very essential, because if this pressure back of the rings were not available, it would be necessary to make the rings sufficiently stiff to insure them against collapse while passing over the ports and when subjected, on their outer surfaces, to the live, or exhaust, steam pressure. Some piston valves are provided with narrow rings and, as far as the balancing is concerned, this is approaching the best theoretical condition, but other conditions have seemed to dictate the use of wider rings. If it is necessary for the rings to travel over the ports, as at present seems to be the case, then because of the necessary springing action of the rings, provided to keep them in contact with the casing, it is essential that bridges should be placed across the ports to carry the rings over; these bridges may be made as broad as convenient, but the same amount of wearing surface cannot be provided in them that is given in the other parts of the casing over which the rings travel; the result, therefore, is that the bridges wear faster than the continuous face of the casing, and in the course of time shoulders are formed over which the rings must travel. This condition prevailing, the ultimate result is that the rings will be broken, with more or less serious consequences, or the casing must be rebored. Efforts to overcome this difficulty have taken the direction of wider rings, and in some designs the rings are made wider than the ports over which they pass in order to insure that the rings cannot drop into the ports. Another reason for using wide rings is as follows: When the very narrow rings were first used it was considered that the ends of the valves were the controling edges, but it was soon found that the edges of the rings were the controling edges. It is necessary to have a shoulder at each end of the valve to retain the rings, and the distance from one end of the valve to the edge of the ring at that end could not be less than ¼ inch with the very narrow ring, so that after the ring begins to uncover the port the steam must be wire drawn around the end of the valve and between the valve and its casing for a distance of ¼ inch. This condition is most serious when the valve is working at a short travel. It was considered necessary to make the edge of the ring correspond with the end of the valve, and the result was a wider ring. Some piston valves at present in use are provided with narrow rings, the outer ones of which are set in from the ends of the valve, and these are reported by those interested in them to be working satisfactorily; but it is believed that practice, as far as the proper distribution of the steam and the providing of the necessary wearing surface are concerned, has shown the desirability of wide rings, whereas the question of properly balancing the valve will require narrow rings, the limiting point of the latter being the discarding of the rings entirely.

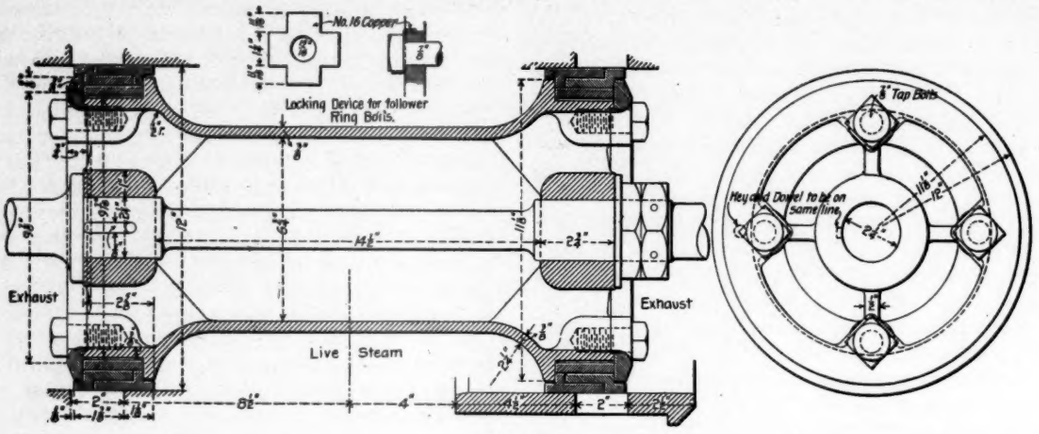


Fig. 3.

Having shown the causes which lead to the use of wide rings, it will be interesting to follow the effects of their use on the balancing of the valve. There is shown in Fig. 3 the section of a piston valve which has been used in a number of locomotives for both freight and passenger service, and which probably illustrates the extreme practice in the use of wide rings. When these valves first went into service it was reported that they were giving entire satisfaction and that the valves were easily operated by the reverse lever, but changes made in the design to provide narrower rings, and reports that it required both engineer and fireman to reverse the engine under steam would indicate otherwise, and the table given below, showing the "load" on the valve, will explain the reason and also substantiate the position taken in this discussion.

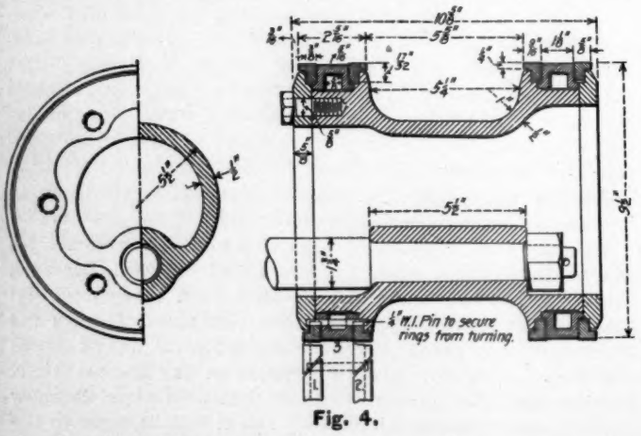


Fig. 4.

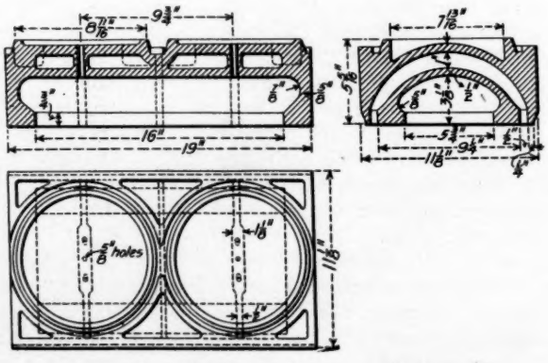


Fig. 5.

Fig. 4 shows the development of moderately wide rings from the narrow rings; originally this valve had narrow rings set in from the ends of the valve, but when it was found that the

edges of the rings were the controling edges the ends of the valve were beveled 3/16 in., as shown, and the rings made L shape, so that the leg would extend to the ends of the valve. The result of this was an improvement in the admission of steam at the expense of the balancing of the valve, but that the latter effect was not serious is shown in the table below.

The number of piston valves is not limited to those shown in these engravings, but as it is the intention merely to call attention to the fact that because a valve is of the piston type does not insure that it is balanced thoroughly and that some piston valves are not as well balanced as the ordinary D valve can be. In Fig. 5 is shown the latter type of valve and the table gives data from which a better appreciation of the value of the balancing of the different types of valves may be obtained.

Location of valve.	Total pressure forcing rings of valve against seat in pounds.; Valves shown in		
	Fig. 3.	Fig. 4.	Fig. 5.
Central (no steam in cylinder)..	32,504	6,460	16,160
Beginning of admission.....	22,894	2,356	9,960
¼ Port opening.....	26,097	1,330	9,714
½ Port opening.....	29,301	1,330	9,478
¾ Port opening.....	29,678	2,356	8,132
Full port opening.....	19,690	3,895	7,382
At cut off.....	32,504	2,356	16,106

The above figures allow for the steam pressure in the steam chest and the steam pressure in the cylinder after the first exhaust, except in first location, marked "Central."

That the use of packing rings on piston valves may seriously affect the balancing is more generally appreciated than it was

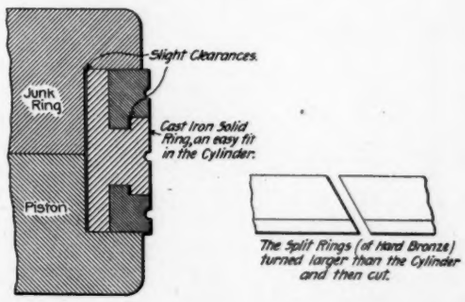


Fig. 6.

a year ago, and efforts have gone so far even as recent attempts to balance the packing rings on the main pistons, and some of the railroads have been approached by inventors who claim to have accomplished such a result; it is extremely doubtful, however, whether the rings can be balanced and fulfill their function regardless of some alleged tests to the contrary and patents issued covering the designs.

[We have taken the liberty to add Fig. 6 to the illustrations presented by Mr. Whyte in order to show a method of making piston packing rings devised by Sir John Durston, recently described in "Engineering." This ring is split and is restricted so that it can open only to a certain extent. The carrier is of cast-iron with two hard bronze rings having lips. A small clearance is allowed between the carrier and the rings, so that the latter may expand up to the clearance allowed. In working, this acts as an ordinary loose ring, but it soon begins to wear the cylinder, and this continues until a perfect fit is obtained. This arrangement is now fitted to all the auxiliary machines in the British navy. Piston valves are necessarily used in all auxiliaries owing to the high steam pressures, and this enhances the importance of a satisfactory piston ring.—Editor.]

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

The 39th semi-annual meeting of the American Society of Mechanical Engineers, held May 9 to 12, in Washington, D. C., was a very successful one. The list of subjects was given last month, and of these the most important will be mentioned. The report of the Committee on the Revision of the Society Code of 1885 for a standard method of conducting boiler trials was presented in final form and is a very important document. The paper on construction and material for pipe flanges brought out a lively discussion, which made it evident that this is a very important subject. Admiral Melville stated that in the war with Spain, steam pipe joints gave more trouble than any other factor about the ships. Mr. Henderson's excellent paper on the manufacture of car wheels was not discussed, because of the lack of members who were sufficiently informed. It was carried over for the next meeting, and it is hoped that it may then receive the attention it deserves. Mr. Henderson showed that this branch of manufacturing was now conducted upon a scientific basis of physical and chemical specification and test, and the desirable composition of wheels was given. The equipment of tall office buildings was an important subject, and the paper by Mr. Bolton gave satisfactory data to show the character and extent of the problems involved. Mr. John Fritz presented a paper describing successful fly wheel designs for rolling mill engines, which included fly wheels which have been running for 25 years without failure. The practice is a valuable precedent for future designers. Prof. Bull's paper on the central heating plant of the University of Wisconsin introduced the subject of central power stations, and the most significant fact brought out in the discussion was that concentration of power was not always advisable, especially when it involved the use of anxiliary elements scattered over a large area, such as pumps for the return of water of condensation. The use of coal handling machinery for comparatively small plants was considered unwise, because unnecessary. The extensive and admirable power plant of Columbia University was described in considerable detail by Mr. E. A. Darling, and the paper is a valuable document because of its completeness. Among the remaining papers the most interesting to our readers was that by Mr. Quereau on the Allen valve for locomotives. The paper states the merits of this valve and it also answers the few objections which have been raised to it. The value of the Allen port in securing good steam distribution was shown by indicator cards. The discussion offered by Mr. Charles T. Porter was chiefly historical. The valve was introduced in England in 1862, and later in this country. The credit for its adoption in general practice in this country was given to Mr. M. N. Forney. The Allen port and the Richardson balancing feature together had made this valve a success.

The desirability of inviting the management of the International Railway Congress to hold the meeting, next after the Paris meeting of 1900, in the United States, was suggested by Col. H. S. Haines before the New York Railroad Club at its May meeting. The idea is an excellent one and it will have the earnest support of all who are interested in bringing about a more perfect understanding among foreign railroad men of American transportation methods.

Gasoline engines have been selected as the most satisfactory source of power for removing the water of drainage from the depressed track work at 16th and Clark Streets in Chicago. The height of the Chicago River varies in such a way as to necessitate pumping the water when at the high levels and yet most of the time natural drainage may be relied upon. The drainage is all directed toward a sump, immediately over which two 8 horse-power Fairbanks-Morse gasoline engine are placed and coupled to centrifugal pumps, which are immersed in the sump. These pumps have a capacity of 2,250 gallons per minute and they are coupled direct to the engines by clutches, which may be thrown into gear after the engines are started. The application of gasoline engines to this service is admirable, because the demand is intermittent. The river is often low enough for several weeks at a time to permit of gravity drainage, and yet the pumping machinery must be ready for instant use when required. This requirement, as well as low cost of operation and small initial outlay required, seemed to be best fulfilled by the gasoline engine.

LOCOMOTIVE DESIGN.—THE WORKING STRENGTH OF MATERIALS.

By F. J. Cole, Mechanical Engineer, Rogers Locomotive Works.

BOILERS.

It is generally conceded at the present time that the normal stress for the plates composing the cylindrical portion, the shell top, dome, etc., in new steel locomotive boilers should be about one-fifth of the ultimate static strength, calculated at the longitudinal riveted joints, which are usually the weakest points. With a properly constructed butt joint, sextuple riveted with double cover strips, holes punched and reamed to size, 85 per cent. of the strength of the solid plate can be readily obtained. A working stress of one-fifth of the ultimate strength may safely be taken as representing good practice, better, no doubt, than the average, with some allowance for possible deterioration in time, caused by corrosion, pitting, grooving and other sources of weakness. It would also permit the pressure to be maintained at its maximum for a greater number of years than is usually possible. This is a point which should be carefully considered in building locomotive boilers, and a sufficient margin of strength allowed over and above the actual amount required when new, so that the maximum efficiency of the engine can be maintained by keeping the boiler pressure at the original figure for which the engine was designed, until the machinery is worn out or other reasons render its retirement from service desirable. An occasional renewal of the firebox flue sheets every few years, and the entire firebox at longer intervals, would be required, as the life of the sheets directly exposed to the fire is necessarily much shorter than that of the outside shell. Too much emphasis cannot be laid upon the fact that because a boiler is entirely safe when new, where the stress is considerably below one-fifth of its ultimate strength, yet in time, owing to corrosion, expansion and contraction, produced by variations in temperature and other weakening causes, the strength may be so reduced that what was originally a safe pressure soon may become a dangerous one. Another side of the case which deserves consideration is the question whether large and small boilers should be designed with the same working stress and the same factor of safety. Inasmuch as corrosion, taking the forms of grooving, pitting, or simply thinning the plate over large areas, is as active in a small boiler as a large one, it would seem a rational policy to design all boilers with a working stress of say one-fourth the ultimate bursting pressure, and then add a uniform amount, say, $\frac{1}{16}$ or $\frac{1}{8}$ inch extra thickness to the plates, to cover the injury caused by improper caulking, which may take place in building or afterward in repairs, the action of corrosive water, rust and other causes. Under these conditions, assuming that the elastic limit is not less than 55 per cent. of the ultimate strength of the plates, with longitudinal joints having an efficiency of 85 per cent., the factor of safety at the elastic limit would be 2.2 at the joint, exclusive of the extra thickness, and 2.6 at the solid plate, where it would be more likely to be attacked by corrosion, as the longitudinal joints are generally above the center of the boiler.

Under the rules of the United States Board of Supervising Inspectors of Steam Vessels, as amended January, 1899, the working stress for boilers built since 1872 is about 26.6 per cent. of the ultimate strength for double-riveted seams, and 27.7 per cent. for single-riveted seams, assuming that the longitudinal joints have an efficiency of 75 per cent. for the first and 60 per cent. in the second case. The rules are:

"Multiply one-sixth of the lowest tensile strength found stamped on any plate in the cylindrical shell by the thickness—expressed in inches or parts of an inch—of the thinnest plate in the same cylindrical shell, and divide by the radius or half diameter—also expressed in inches—and the sum will be the pressure allowable per square inch of surface for single riveting, to which add 20 per cent. for double riveting, when all the rivet holes in the shell of such boilers have been 'fairly drilled' and no part of such hole has been punched.

"Where butt straps are used in the construction of marine boilers the straps for single butt strapping shall in no case be less than the thickness of the shell plates; and when double butt straps are used the thickness of each shall in no case be less than five-eighths of the thickness of the shell plates.

"The hydrostatic pressure applied must be in the proportion of 150 pounds to the square inch to 100 pounds to the square inch of the steam pressure allowed."

The working pressures obtained by means of the foregoing rules do not vary according to the strength or style of the joints, except as single or double riveted, nor do the rules provide what is considered at the present time a sufficient factor

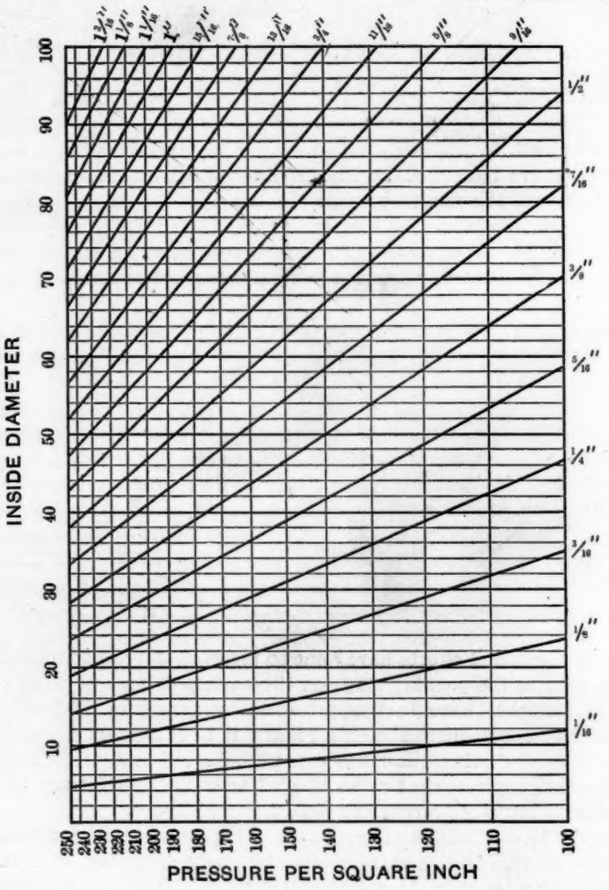


Fig. 1.

of safety for locomotive boilers working at high pressures. As the factor is only about 3.9 of the ultimate bursting pressure, the rules may be regarded as unsatisfactory and unsuitable for modern locomotive practice. In adjusting the pressures on boilers built several years ago which carry moderate pressures, they may be found useful.

In Lloyds rules (British) for the strength of cylindrical boiler shells the working pressure is calculated from the following formula:

$$\frac{C \times T \times B}{D} = \text{working pressure.}$$

Where C = coefficient as given in table below.
T = thickness of plate.
D = mean diameter of shell.
B = percentage of strength of joint found as follows, the least percentage to be taken:

$$\text{For plate at joint } B = \frac{P - d}{P} \times 100$$

$$B = \frac{n \times a}{P \times T} \times 85 \text{ with steel rivets in steel plates.}$$

$$B = \frac{n \times a}{P \times T} \times 70$$ with iron rivets in steel plates.

(In case of rivets being in double shear, 1.75 a is to be used instead of a) where P = pitch of rivets; d = diameter of rivets; a = sectional area of rivets; n = number of rows of rivets.

Coefficient "C" for Steel Boilers.

Description of longitudinal joint.	For plates ⅜ thick and under.	For plates over ⅜ thick and above ⅝	For plates over ⅝ thick and above ¾.	For plates above ¾ thick.
Lap joints	200	215	230	240
Double butt strap joints	215	230	250	260

This is based broadly upon a factor of safety of five with certain limitations and additions. The working stress per square inch of section is as follows:

Iron Boilers.

	Plates ½-in. and less.	Plates ⅝-in. and above ½-in.	Plates over ¾-in.
Lap joint, punched holes	7,750	8,250	8,500
Lap joint, drilled holes	8,500	9,000	9,500
Double butt strap joint, punched holes	8,500	9,000	9,500
Double butt strap joint, drilled holes	9,000	9,500	10,000

Steel Boilers.

	Plates ⅝ in. and under.	Plates ¾-in. and ⅞-in.	Plates ⅞-in. and above 1-in.	Plates over ¾-in.
Lap joints	10,000	10,450	11,500	12,000
Double butt strap joints	10,450	11,500	12,500	13,000

The following extract from a useful little table printed under the caption, "Safe Working Pressures for New Boilers," in "The Locomotive," a paper published under the auspices of The Hartford Steam Boiler Inspection Co., is worthy of record:

"The following short table, which we use in our own practice in fixing the working pressure of new boilers, may be of use to our readers. The pressures given by the table are not as high as are allowed by the rules of the United States Board of Supervising Inspectors, but those, in our opinion, are somewhat greater than can be safely carried after a boiler has been in use for some time."

Steel of 60,000 Pounds per Square Inch of Section.

Thickness of plates.	Diam. of rivets.	Pitch double riveting.	Safe working pressure in pounds per square inch for shells with longitudinal seams double riveted. 36 Diam.	42 Diam.	48 Diam.	54 Diam.	60 Diam.	66 Diam.	72 Diam.
¼	⅝	2⅜	135	115	100	85	...	...	...
5/16	¾	2¾	160	135	120	105	95	...	...
⅜	⅞	3¼	190	160	140	125	115	100	95
7/16	1	3⅜	220	185	160	145	130	120	110
½	⅛	3½	250	205	180	165	150	135	125

[For thicker plates the pressures can be obtained approximately by simple proportion.—F. J. Cole.]

The Hartford Steam Boiler Inspection Co. says: "We consider 5 to be the best factor of safety when all things are considered, though we sometimes allow 4½ when the workmanship is known to be first class, and the materials of which the boiler is made have been carefully selected and tested."

Nominal Factors of Safety.

Lloyds (British), about 4.5 to 5.5.

Board of Trade (British), 4.5 for best construction and workmanship.

German Lloyds, 5 to 4.65, according to thickness of plates.

Laws of City of Philadelphia, 5.

The strength, thickness of plates and working pressure for circular boiler shells may be determined by the following formulae:

Let t = thickness of plates.
P = pressure per square inch.
F = factor of safety.
D = inside diameter.
J = percentage of strength of joint to solid plate.
S = lowest tensile strength of plates per square inch.
M = stress per square inch.

$$\text{Then } t = \frac{D\,P\,F}{2\,S\,J} = \frac{D\,P}{2\,M\,J}$$

$$P = \frac{S\,t\,2\,J}{D\,F} = \frac{M\,2\,J\,t}{D}$$

$$F = \frac{S\,t\,2\,J}{D\,P} \qquad M = \frac{P\,D}{2\,t\,J}$$

The value of S and t combined may be taken directly from Table No. 1.

Example: A boiler, 65 inches greatest inside diameter, steam pressure 200 pounds per square inch; joint efficiency 85 per cent. of solid plate; lowest tensile strength of plate 55,000 pounds; factor of safety 5; required the thickness of plates.

$$t = \frac{D\,P\,F}{2\,S\,J} = \frac{65 \times 200 \times 5}{2 \times 55000 \times .85} = \frac{6500}{9350} = .695$$ or about $\frac{11}{16}$ inch. Or if the thickness is known to find the pressure (otherwise the same as the preceding example).

$$P = \frac{S\,2\,J\,t}{D\,F} = \frac{55000 \times 2 \times .85 \times .695}{65 \times 5} = 200 \text{ lbs.}$$

TABLE NO. 1.

Strength of Steel Boiler Plates, 1 Inch Wide.

Thickness.	Tensile strength. 50,000 lbs.	55,000 lbs.	60,000 lbs.
¼	12,500	13,750	15,000
9/32	14,062	15,469	16,895
5/16	15,625	17,188	18,750
11/32	17,187	18,907	20,625
⅜	18,750	20,625	22,500
13/32	20,312	22,344	24,375
7/16	21,875	24,063	26,250
15/32	23,437	25,782	28,125
½	25,000	27,500	30,000
17/32	26,562	29,219	31,875
9/16	28,125	30,938	33,750
19/32	29,687	32,657	35,625
5/8	31,250	34,375	37,500
21/32	32,312	36,094	39,375
11/16	34,375	37,813	41,250
23/32	35,937	39,532	43,125
¾	37,500	41,250	45,000
25/32	39,062	42,969	46,875
13/16	40,625	44,687	48,750
27/32	42,187	46,406	50,625
⅞	43,750	48,125	52,500
29/32	45,312	49,844	54,375
15/16	46,875	51,562	56,250
31/32	48,437	53,281	58,125
1	50,000	55,000	60,000

By means of diagram Fig. 1 the proper thickness of plates for pressures from 100 to 250 pounds and sizes of boilers up to 100 inches diameter may be obtained without any calculation.

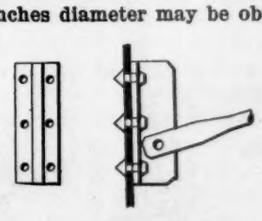


Fig. 2.

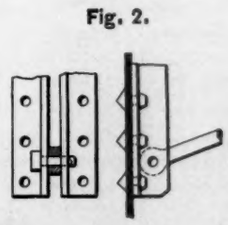


Fig. 4.

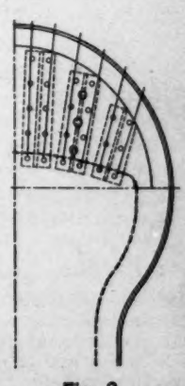


Fig. 3

It is based upon a factor of 5, or upon a stress of 11,000 pounds per square inch, assuming the minimum ultimate tensile strength of the plates to be 55,000 pounds and the joint efficiency 85 per cent. Example: What thickness of plate is required for a boiler 70 inches inside diameter, working pressure 200 pounds. Follow the horizontal line marked 70 until it intersects the vertical line marked 200, the diagonal line marked ¾ at the intersection is the thickness required.

TABLE NO. 2.
Boiler Steel.

Name.	Kind.	Desired.		Length of spec.	Will reject.			Chemistry.															
					Tensile strength.			Carbon.			Phos.			Man.			Sil.			Sul.			Cop.	
		Tensile strength	Elong'n		Max.	Min.	Elongation.	Desired.	Max.	Min.	Desired.	Max.	Min.	Desired.	Max.	Min.	Desired.	Max.	Min.	Desired.	Max.	Min.	Desired.	Max.	Min.
Master Mechanics' Association	Shell	60,000	25%	8″	65,000	55,000	20	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..
	Firebox	60,000	28	8″	65,000	55,000	22	.18	.25	.15	.03	.035	..	.4	.45	..	.02	.03	..	.02	.035	..	..	..	..
Penna. R. R. Co., 1892	Shell	60,000	25	8″	65,000	55,000	20	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..
	Firebox	60,000	28	8″	65,000	55,000	22	.18	.25	.15	.03	.04	..	.4	.55	..	.02	.04	..	.02	.05	..	.03	.05	..
Erie R. R.	Shell	60,000	25	8″	65,000	55,000	20																
	Firebox	60,000	28	8″	65,000	55,000	22																
C. B. Q. R. R. Co.	Shell	60,000	28	8″	65,000	55,000	22	.18	.25	.15	.03	.035	..	.4	.45	..	.02	.03	..	.02	.035	..	..	..	..
	Firebox	60,000	28	8″	65,000	55,000	22	.18	.25	.15	.03	.035	..	.4	.45	..	.02	.03	..	.02	.035	..	..	..	..
Baldwin Locomotive Works	Shell	60,000	25	8″	65,000	55,000	20	..	.25	.15	..	.05	..	..	.45	..	..	.03	..	..	.05	..	.	..	.
	Firebox	60,000	25	8″	65,000	55,000	20	.	.25	.15	..	.03	..	..	.45	..	..	.03	..	..	.035	..		.	
Rogers Locomotive Co.	Shell	60,000	25	8″	65,000	55,000	20																
	Firebox	54,000	28	8″	58,000	50,000	22	.18	.25	.15	03	.035	..	.40	.45	..	.02	.03	..	.02	.035	..	..	..	..
United States Navy	Shell	.....		8″	67,000	58,000	25	..	..	..	..	.035	..	..	..	..	..	..	..	..	.04	..	..	..	..
	Firebox	...	....	8″	58,000	50,000	26	..	..	..	..	.035	..	..	..	..	..	..	..	..	.04	..	..	..	..
British Admiralty	.....	.....	.....	8″	67,200	58,240	20																
U. S. Navy, cruisers, 1887.	Shell	.....	.....	8″	63,000	57,000	25																
	Firebox	.....	.. ..	8″	55,000	50,000	29																

Table No. 2 gives the physical and chemical requirements of a number of specifications for boiler shell and firebox steel. It includes those of a number of prominent railroads, including those of the American Railway Master Mechanics' Association, the United States Navy, etc. With the exception of the British Admiralty and the United States Navy, the minimum tensile strength specified for flange or shell steel is 55,000 pounds per square inch. A working stress of 11,000 pounds for a factor of 5 and of 12,220 pounds for a factor of 4.5 seems most suitable for average conditions.

Heavy steel tees are generally riveted to the flat surface of the front and back heads, to which braces are secured. The usual method is shown in Figure 2, where the brace terminates in a jaw and is secured to the tee by means of a bolt and nut. The diameter of these bolts suitable for different sizes of braces calculated for a single shearing stress of 7,500 pounds per square inch of area, and in double shear of 1.75 × 7,500 may be obtained from table No. 3.

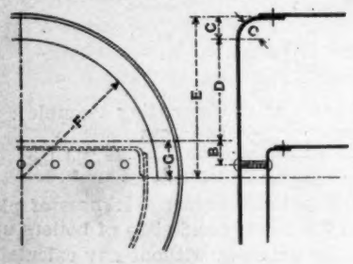


Fig. 5.

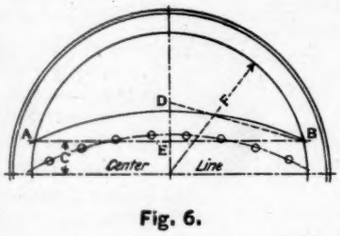


Fig. 6.

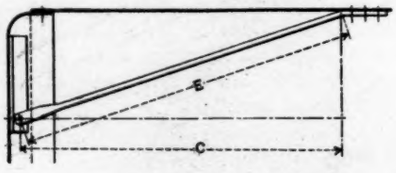


Fig. 7.

TABLE NO. 3.

Diam. of brace.	Area.	Ultimate strength at 48,000 lbs.	Safe working stress. 8000	9000	Size and number of rivets in foot.	Size of foot.	Diam. of pins, double shear, 7500 x 1.75.
⅞	.60	28800	4800	5400	2—⅞	½″ x 2″	⅝″
1″	.78	37500	6250	7020	2—⅞	⅝″ x 2¼″	⅝″
1⅛	.99	47500	7920	8910	2—⅞	⅝″ x 2¼	1″
1¼	1.22	58560	9760	10980	2—1⅛	¾″ x 2¾″	1″
1⅜	1.48	71040	11840	13320	2—1⅛ or 2—⅞	⅞″ x 2¾″	1⅛″
1½	1.77	84960	14160	15930	2—1⅛ or 2—1⅛	1″ x 3″	1¼″

The tees should not be less than 5 in. wide to allow room for the rivets and about 4 to 4¾ inches in depth. Figure 3 shows the usual arrangement of tees when used in radial stay boilers. In Belpaire and crown bar boilers the tees would be arranged vertically instead of radially. Heavy steel angles about 4x4 inches are often used in place of tees for attaching the ends of braces to flat sheets. They are placed back to back, with a sufficient space between to admit the ends of the braces as shown in Fig. 4, the ends being forged solid instead of using the jaw connection. The rivets should be about 4 to 4½ inches apart lengthwise of the angles or tees, the usual diameters of the rivets being ¾, ⅞ and 1 inch, varying according to the thickness of the plates and attachments.

Three-quarter inch rivets are used for plates not exceeding ⅜-inch; ⅞-inch rivets for plates from 7/16 to ½-inch; and 1-inch rivets for plates 9/16-inch and over. Crowfeet, drop forged out of solid metal without welding, are used for attaching the braces to the boiler shell in small corners where there is not sufficient room for the tee iron.

The braces used for strengthening the flat surfaces, such as the upper parts of the back and front heads, should be designed for a working stress of 8,000 to 9,000 pounds per square inch. The stiffness of the head itself should be disregarded and the braces proportioned to carry the entire load. The area to be supported should be taken at 2 inches from the center of the top row of staybolts, Figure 5, or half their distance apart to the edge of the round corners, or to the center of radius of the corner.

The area to be braced, of the back head flat surface of a boiler having a flat crown sheet, is:

$A = F^2. 7854 - (G\ F)$

When A = area of flat surface requiring bracing.
B = radius of corner.
B = ½ the distance apart of the staybolts.
S = maximum working steam pressure.
D = height of flat surface.
E = radius of back head.
F = radius of flat surface.
G = center of staybolt to center of back head + B.

AS is the total force to be resisted by bracing.

$\frac{AS}{8{,}000 \text{ to } 9{,}000}$ = the number of square inches of iron required in the braces.

When the crown sheet is curved for radial stay boilers, the area of the segment which requires bracing is shown in Figure 6. Draw the line, D B, so that equal areas will be left inside and outside the curved line. Then the area of the segment will equal one-half the area of the circle, having the radius, F, less the rectangle, 2 F C, and the triangle, D, E, B. If the braces are attached to the shell at a considerable angle, as in Figure 7, due allowance should be made. Counting the normal tension on the brace (if it were at right angles to the back head) at 1, then the actual tension will = $\frac{E}{C}$.

Assuming that the minimum strength of the best wrought iron suitable for boiler bracing is 48,000 pounds per square inch, and that 37,500 pounds is the average resistance of rivet iron in single shears, then the factor of safety at 9,000 pounds is 5.33, and at 8,000 pounds it is 6. The rivets attaching the foot to the boiler shell may safely be proportioned for 7,500 pounds per square inch in single shear or a factor of 5.

A brace 1 inch diameter would have an ultimate breaking strength of 37,700 pounds, and a safe working strength of 6,250 to 7,020 pounds. It should be attached to the T iron, angle iron or foot by a pin not less than ⅞-inch diameter if in double shear, and by not less than 2 rivets driven in 13/16 holes if in single shear. Table No. 3 will be found useful in proportioning the braces and pins.

SOME PRINCIPLES GOVERNING THE PROPER DISTRIBUTION OF LOCOMOTIVES IN ECONOMICAL RAILWAY OPERATION.

By E. M. Herr.

Like almost all important matters in railroad operation, the distribution of power requires the collaboration and close co-operation of several officers. The heads of both the mechanical and operating departments must frequently consult and supplement each other's knowledge before the distribution of power can be intelligently handled. When this is done, however, it is believed the proper officer to make the distribution is the head of the mechanical department, as it is easier for him to obtain exact information relative to the operating conditions than to give information as to mechanical questions, owing to their necessarily technical character.

The distribution as well as maintenance of locomotives being under the head of the mechanical department tends toward better maintenance and more satisfactory performance through less engine failures with their attendant expense and interference with successful and economical operation. In order to make the head of the mechanical department responsible for economical operation, as well as maintenance of locomotives, the cost of locomotive repairs should not be judged, as is usual, on the basis of the engine miles run only, but also and principally by the cost per ton-mile hauled. These statistics, if used to judge the economy with which locomotives are maintained, give the responsible mechanical officer an especial interest, as they should, in the real problem of economic railroad operation, the reduction of the cost per ton mile. The weight and horse-power capacity of locomotices used in main line service, provided the density of the traffic is the same, should be such as to enable if possible the same weight of train to be moved over the entire length of the line, thus avoiding the necessity of breaking up through trains at division points.

When the through trains are a large proportion of the total business on any division, the above distribution of power is undoubtedly the most economical. When, however, a considerable portion of the business originates and leaves a division at some point on it or at its terminii, thus making such business local thereto, heavier power than the above plan would call for can be advantageously assigned, owing to the possibility of increasing the average train load on such divisions with little or no increase in switching expense.

The number of locomotives to be assigned to any division is largely governed by the uniformity of the traffic movement. When this is about constant in volume in both directions, whether of equal volume or not, less power need be assigned than when the volume averages the same in each direction but varies considerably from day to day. In the latter case the surplus power is necessary to avoid running power light in one direction one day and perhaps in the opposite direction the next. When this condition prevails, however, a careful study of the situation must be made, as it may easily prove more economical to run power light occasionally than to decrease too much the average mileage made per locomotive by largely increasing the assignment of power. The character of the water supply may, if very bad, require a larger assignment of power due to an increase in boiler repairs and consequent laying up of power otherwise available.

The importance of a careful study of the actual conditions of traffic movement in order to intelligently distribute power cannot be too strongly emphasized. This pertains not only to a study of the local conditions of movement on each division, but what is still more important, an accurate knowledge of the periodic fluctuations of business on different parts of the system and a careful preparation for the prompt reassignment of power adapted to handle the increased traffic in one locality by moving such power from the place where business is decreasing. Such fluctuations occur on large railways in almost all parts of the country, and are generally due to the opening and close of navigation on lakes and rivers, the movement of a new harvest of grain, or fruit, etc.

On some roads a certain assignment of power is made to each division, which is supplemented in a haphazard way when the traffic is heaviest and is partly laid up when the movement is least. On many of our larger roads the heaviest traffic does not come on more than one part of the line at a time. It may be, for example, heavy on the northern or eastern divisions in the Spring and on the southern or western in the Fall. Such a condition gives an opportunity for economical operation by the prompt redistribution of power now not often fully utilized. It is, of course, common to send power from points where it is not needed to points where it is in demand, but generally without due regard to the type of locomotives best adapted to the service, on account of the reassignment having to be made hurriedly or without previous preparation.

The greatest economy of operation not only requires that sufficient locomotive power shall be available to meet the requirements of traffic on any division, but this power should also be the most efficient obtainable. Heavy locomotives, if available, should be moved from some distance, making them work their way in transit, if they can be used to advantage where power is needed, rather than transfer lighter power simply because it is perhaps on the adjoining division and convenient.

In assigning power, care must be used to discriminate between the power requirements of different divisions having perhaps the same ruling grades but different kind of grades or with the same degree and kind of grades with different conditions of operation.

On division A the grades may be short and constantly changing, giving the profile a sawtooth appearance. On division B the same steepness of grade may be practically continuous for long distances. Both these divisions having the same steepness of grade might at first thought be regarded as requiring the same kind of locomotives. The requirements are, in fact, quite different. As far as tractive power is concerned, the grades being equally steep the requirements are the same, calling for locomotives of equal weight and cylinder power. The difference comes in the requirement for boiler power or, what practically amounts to the same thing, the horse-power capacity of the locomotive. Locomotives were formerly built with more consideration for the mere weight and cylinder capacity or tractive power, and less for the boiler or steam generating capacity, than at present. Such locomotives are much better adapted to division A, and locomotives with larger boilers and better steam generating or horse-power capacity to division B.

In order to distribute power properly, it must be available when required. During seasons of light business, owing to an effort to keep the net earnings as uniform as possible, the shop and round-house forces are reduced or shorter hours worked, thus making it impossible to do work of repairs when locomotives can best be spared for shopping. This is generally short-sighted policy and poor economy. It also makes the intelligent and efficient distribution of power more difficult and often impossible, as the heavy first-class locomotives, instead of being shopped when least needed for work, are often kept out of the shops during the dull seasons on account of the decreased force being unable to turn as many of them out as of lighter engines and the immediate need being for more locomotives of any kind rather than more efficient locomotives.

For best economy the prevailing practice of reducing shop forces and expenses during periods of dull business should undoubtedly be reversed. The shops should be run full when business is light and locomotives available for shopping, and shop forces and expenses reduced when business is heavy and the power is needed in service.

Many mechanical officers will object to transferring power periodically, on account of the tendency to increase the cost of repairs, as locomotives are brought from the control of one master mechanic to that of another. This is largely, if not entirely, similar to the old argument that each engineer must run his own engine or repairs would be increased, and will soon be solved in the same way.

Good distribution of power will not alone accomplish the best

results; it must be accompanied and supplemented by good operation, both at the hands of the train-master, chief dispatcher and master mechanic also; but no amount of skill on the part of these officials will enable the most economical results to be obtained if the proper power is not available. For the proper distribution of power we therefore have the following:

Study carefully both grades and traffic conditions on each division.

Assign power so that traffic from one division to the next can be handled with the least switching.

Transfer power to meet periodic fluctuations in traffic by a well-developed plan, having due regard to doing the heavy repairs, as far as possible, during the dull season.

Assign locomotives of small boiler capacity to divisions having short grades or on which speed is slowest, other things being equal, and largest boiler and horse-power capacity locomotives on long, continuous grade divisions or where maintenance of speed is more important.

Co-operate heartily and confer frequently with officers of the operating department as to the requirements of the service and the conditions of traffic.

ECONOMICS OF FREIGHT LOCOMOTIVE OPERATION.

By G. R. Henderson.

Assistant Mechanical Engineer, Schenectady Locomotive Works.

In these days of railroad competition, no subject interests the operating departments as much as the economical handling of the freight traffic, especially upon roads where freight forms the larger part of the business. As the fuel bills of any large railroad are a very considerable proportion of its operating expenses, the relative consumption of coal per ton mile is considered of great importance, but there are often other factors of operation which quite overcome and sometimes reverse the final figures, when fully considered.

With the present tendency to load locomotives up to the maximum limit, managers are loath to believe that such a result is accompanied by an increased fuel consumption per ton-mile hauled back of the tender, and it will, perhaps, surprise some to know that under certain conditions the increase is very great.

In order to determine what this variation would amount to in actual practice, calculations were made on a locomotive of the consolidation type, with 20 by 24-inch cylinders, 150 lbs. boiler pressure, 50-inch drivers, and having a total weight of engine and tender of 105 tons. The heating surface was 1,774 square feet, and the grate area 31 square feet. The maximum cut-off permitted by the valve gear was 90 per cent. of the stroke, and as the most economical cut-off for a single expansion engine is at 25 or 30 per cent., deductions were made at 30, 60 and 90 per cent. cut-offs, at speeds of 5 and 10 miles per hour. As the boiler was not large enough to supply steam at the 90 per cent. cut-off at a higher speed than 10 miles per hour, the computations for 15 miles an hour were made for 30, 45 and 60 per cent. cut-offs. For each of the three speeds we have, therefore, determined the consumption of fuel at the point of greatest fuel economy, the point of greatest power which the engine can exert at the given speed, and at a point intermediate between the two. The coal consumption for each point was calculated from the cylinder volume at point of cut-off, the cut-off pressure, mean effective pressure and weight of steam, allowing 25 per cent. for cylinder condensation at 30 per cent. cut-off, and 10 per cent. at 90 per cent. cut-off. The rate of combustion and corresponding rate of evaporation were also figured, and also the tractive power of the engine and the load which it would haul (back of tender) on a level, on a one per cent. grade and on a two per cent. grade. From these data the pounds of coal per 100-ton-miles were readily computed. The following table explains the method followed, giving the values at 10 miles per hour:

Data figured at.....	30	60	90 per ct. cut off.
Cylinder volume	1.3	2.6	3.9 cubic feet.
Cut-off pressure	106	114	136 lbs. per sq. in.
Mean effective pressure....	66	100	130 lbs. per sq. in.
Mean available pressure ...	61	92	120 lbs. per sq. in.
Weight per cu. ft. steam..	.275	.295	.340 lbs.
Allowance for condensation	25	15	10 per cent.
Steam used per hour.....	9,009	17,781	29,069 lbs. from and at 212°.
Available tractive force	11,712	17,664	23,040 lbs.
Coal burned per hour.....	900	2,223	4,457 lbs.
Rate of combustion.....	30	72	144 lbs. per sq. ft. grate.
Rate of combustion.....	.5	1.2	2.5 lbs. per sq. ft. H. S.
Evaporative rate	10	8	6.5 lbs. water per lb. coal from and at 212°.
Load Hauled, Gross.			
On level (5½ lbs. resistance).....	2,135	3,205	4,180 tons.
On 1% grade (25½ lbs. resistance.....	460	692	903 tons.
On 2% grade (45½ lbs. resistance).....	257	388	505 tons.
Load Hauled, Net.			
On level	2,030	3,100	4,075 tons.
On 1% grade	355	587	798 tons.
On 2% grade	152	283	400 tons.
Pounds Coal per 100 Ton-Miles, Net.			
On level	5	7	11 lbs.
On 1% grade	25	38	56 lbs.
On 2% grade	59	79	112 lbs.

The various values given in the table were obtained by the method described in the "Report of Committee on Ratios of Grate Area, Heating Surface and Cylinder Volume" to the Master Mechanics' Association, 1897.

Diagram No. 1 shows graphically the results of calculations on this type of locomotive at speeds of 5, 10 and 15 miles per hour, and on levels 1 per cent., and 2 per cent. grades. The abscissae represent tons of train hauled (back of tender), and the ordinates the pounds of coal consumed per 100 ton-miles (also back of tender). The three small circles on each curve represent the points calculated at the different proportions of cut-off, corresponding to maximum load, most economical load, and an intermediate loading. The farthest point to the right is the greatest load which can be drawn by the engine under consideration (theoretically) on the grade and at the speed assumed.

This diagram shows that the most economical loading, as far as fuel alone is concerned, is that suited to the early cut-off, and that increasing the load means an increase in the amount of fuel per 100 ton-miles. On grades the variation is more striking than on levels. For instance, while on a level, at a speed of ten miles an hour, a load of 2,030 tons consumes 5 lbs. of coal per 100 ton-miles, and a load of 4,075 tons, 11 lbs. per 100 ton-miles, on a 2 per cent. grade, a load of 152 tons requires 59 lbs. of coal, and a load of 400 tons, 112 lbs. of coal per 100 ton-miles. These figures should not be confused and considered as merely the increase in consumption following an increase in the load, but it should be noted that the increase is in units of 100 ton-miles, or for the same amount of

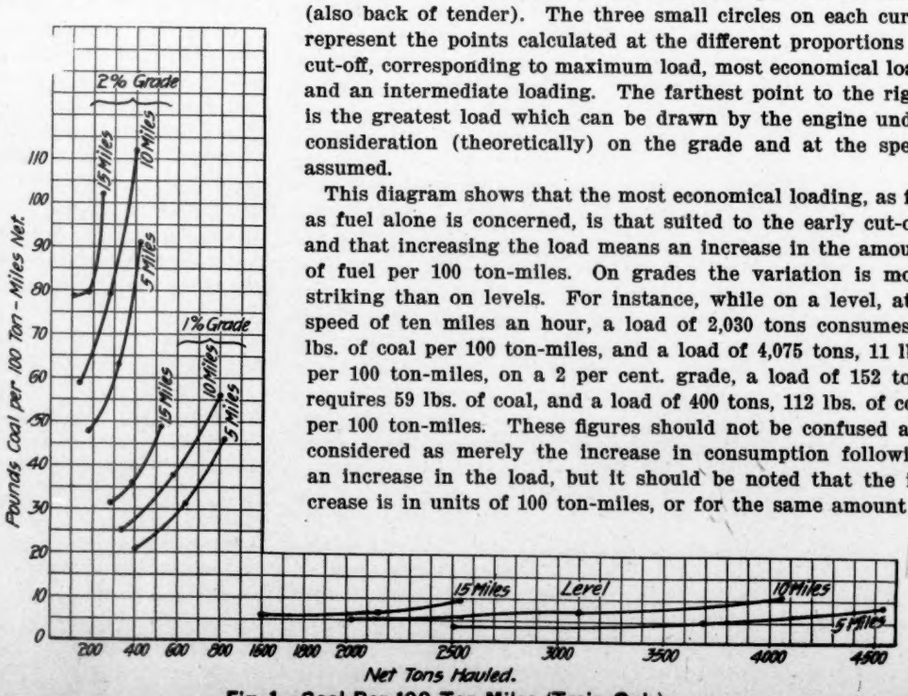


Fig. 1.—Coal Per 100 Ton Miles (Train Only).

work. This chart shows at a glance how unsatisfactory any comparisons of fuel consumption of locomotives are likely to be even on a ton-mile basis, except when the grade of the division and the working conditions are known and are subject to comparison. But, as above suggested, if the heavy train loads mean a loss in economy as far as fuel is concerned, it does not necessarily follow that the total cost of operation will be greater. In order to arrive at the total cost of train operation, as far as it is affected by the expenses directly chargeable to the individual trains, but not including maintenance of track, stations, signals, etc., which are approximately constant, regardless of the weight of trains, we will proceed further with our calculations, based upon the following items of expense:

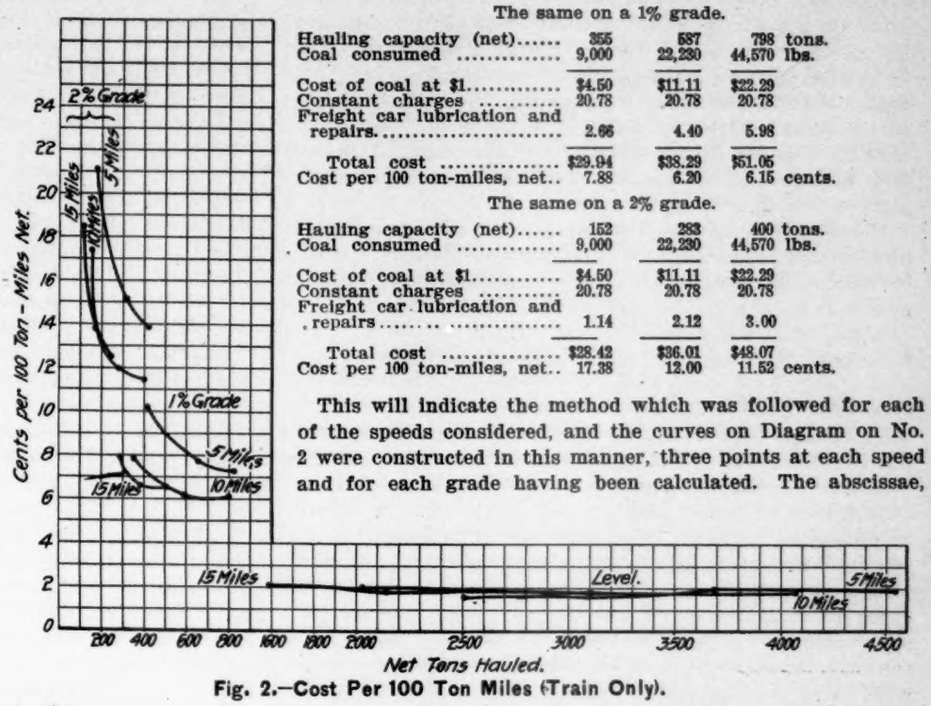


Fig. 2.—Cost Per 100 Ton Miles (Train Only).

Coal consumed	at $1.00	per ton.
Locomotive repairs	at .05	per mile.
Locomotive lubrication	at .25	per hundred miles.
Sand, tools and supplies	at .13	per hundred miles.
Hostlering, etc	at .90	per hundred miles.
Engineman	at .40	per hour.
Fireman	at .20	per hour.
Conductor	at .30	per hour.
Brakeman	at .15	per hour.
Freight car repairs	at 2.60	per thousand car miles.
Freight car lubrication	at .05	per thousand car miles.
Assumed weight of each car loaded.	35	tons.

For interest and depreciation on the locomotive we allow as follows:

Depreciation at 10% on $10,000	$1,000 per year
Interest at 5% on $10,000	500 per year
Total	$1,500 per year

If we allow 300 working days in a year, we have 1,500 ÷ 300 = $5.00 per day of 24 hours, or, figuring on ten-hour runs, and two hours' lay over we have per trip $2.50.

We can now figure the charges which are constant for a trip of 100 miles, or 10 hours, at 10 miles per hour, regardless of the train load, as shown below:

Locomotive repairs at $.05 per mile		$5.00
Locomotive lubrication at .25 per 100 miles		.25
Locomotive sand, tools and supplies at .13 per 100 miles		.13
Hostlering, etc., at .90 per 100 miles		.90
Interest and depreciation for ½ day		2.50
Wages of crew for 10 hours:		
Engineman	$4.00	
Fireman	2.00	
Conductor	3.00	
Two brakemen	3.00	
		12.00
Total constant charge		$20.78

We may estimate freight car lubrication and repairs as follows:

Repairs per 1,000 car miles	$2.60
Lubrication per 1,000 car miles	.05
Total per 1,000 car miles	$2.65
Cost per car mile	$0.00265
At assumed weight of loaded car, 35 tons, we have	
Cost per ton-mile	$0.000075

We are now prepared to figure out the three points on each of the three curves for a speed of 10 miles per hour.

Train cost, at 10 miles per hour for a 10-hour run, or 100-mile trip, on a level:

Data.	30	60	90	per ct. cut-off.
Hauling capacity (net)	2,030	3,100	4,075	tons.
Coal consumed	9,000	22,230	44,570	lbs.
Cost of coal at $1 per ton	$4.50	$11.11	$22.29	
Constant charges	20.78	20.78	20.78	
Freight car lubrication and repairs	15.22	23.21	30.50	
Total cost	$42.50	$57.10	$75.57	
Cost per 100 ton-miles, net	2.00	1.78	1.81	cents.

The same on a 1% grade.

Hauling capacity (net)	355	587	798	tons.
Coal consumed	9,000	22,230	44,570	lbs.
Cost of coal at $1	$4.50	$11.11	$22.29	
Constant charges	20.78	20.78	20.78	
Freight car lubrication and repairs	2.66	4.40	5.98	
Total cost	$29.94	$38.29	$51.05	
Cost per 100 ton-miles, net	7.88	6.20	6.15	cents.

The same on a 2% grade.

Hauling capacity (net)	152	283	400	tons.
Coal consumed	9,000	22,230	44,570	lbs.
Cost of coal at $1	$4.50	$11.11	$22.29	
Constant charges	20.78	20.78	20.78	
Freight car lubrication and repairs	1.14	2.12	3.00	
Total cost	$28.42	$36.01	$48.07	
Cost per 100 ton-miles, net	17.38	12.00	11.52	cents.

This will indicate the method which was followed for each of the speeds considered, and the curves on Diagram on No. 2 were constructed in this manner, three points at each speed and for each grade having been calculated. The abscissae, as in Diagram 1, represent the total load hauled back of the tender, while the ordinates indicate the cost in cents per 100 ton-miles. It will be seen at a glance that the total cost on the economical rate is entirely different from that due to coal consumption only. Of course this only holds strictly true for the prices and wages stated above, but the method of making the deductions would be the same, the different values being changed to suit the conditions being considered. The following remarks must not be understood, therefore, to apply only to such cases as may be covered by the figures assumed.

At a given speed the greatest economy is shown when the engine is loaded to the full limit of its capacity. This is illustrated by the fact that nearly all the curves have the lowest cost at a point agreeing with the maximum power of the engine, which is the right hand end of the various curves. This reverses the opinion formed by the coal consumption charge, and shows that the other features more than balance the loss in economy in the coal consumption.

The curves demonstrate that 10 miles per hour with the full load limit of the engine is about the cheapest speed at which to operate, this being particularly so on grades. Thus, on a 2 per cent. grade, we find that the engine, loaded to its maximum power, can haul freight at a cost of 11.52 cents per 100 ton-miles, whereas at 15 miles per hour the cost is 12.51 cents, and at 5 miles an hour, 13.84 cents per 100 ton-miles. On a 1 per cent. grade the difference is not so great, but is in the same direction. We would, therefore, conclude that for the conditions and prices assumed, 10 miles per hour is by far the most economical speed at which to operate freight trains on grades.

For a given load, 15 miles per hour is cheaper than 5 or 10 miles, and 10 miles is cheaper than 5 miles, which would indicate that if the engine is loaded to pass a 2 per cent. grade at 10 miles per hour, which, in this case, would mean a train load of 400 tons, on a 1 per cent. grade, it would be economical to increase the speed to 15 miles per hour, assuming that the load back of tender had remained constant.

On a given grade, we find that a very small reduction in the tonnage (when at the limit of power of the engine) allows the speed to be increased from 5 to 10 miles an hour, and at the same time effects a large reduction in the cost per 100 ton-miles. Take, for instance, at 5 miles per hour, where we find that the locomotive under consideration can haul 413 tons on a

2 per cent. grade, at a cost of 13.84 cents per 100 ton-miles. By reducing the load to 400 tons the speed can be increased to 10 miles per hour, and the cost of hauling will be only 12.52 cents per 100 ton-miles, or by decreasing the load about 4 per cent. we are not only able to double the speed, but also reduce the cost per 100 ton-miles about 15 per cent. The same facts hold good on a 1 per cent. grade in about the same proportion.

This is often overlooked in practical operation, but it has been exemplified in actual work, and goes to demonstrate that overloading to such an extent as to reduce the speed of the engine is not always an economical method of operation. A case known to the writer was that in reducing the load on a locomotive from 650 to 610 tons, the speed was increased from 5 to 10 miles per hour, reducing considerably the cost of operation. This must be understood, of course, as applying only to the maximum load at each speed, as a reduction of load without an increase in speed would cause additional expense, as illustrated by Diagram No. 2.

On a level section of track we find that there is practically no difference in the cost, no matter whether the speed be 5 or 15 miles per hour, or whether the engines be heavily or lightly loaded, as in this case the difference in coal consumption balances the other charges.

It would seem from the above that for the conditions assumed, 10 miles is the most economical speed at which to operate on parts of roads which are not level, but contain grades more or less heavy, but that on level stretches the speed can be varied without much effect on economy.

EXHAUST AND DRAFT APPLIANCES.

The subject of exhaust and draft appliances of locomotives, to be presented to the International Railway Congress at its Paris session next year, has been intrusted to Mr. C. H. Quereau, Master Mechanic of Denver & Rio Grande Railroad, as reporter. A circular of inquiry embracing an admirable series of questions for the purpose of bringing out information as to present practice has been received, and in view of the fact that this subject is of great importance and very little understood, it is to be hoped that the answers will be thoughtfully and carefully given. Notwithstanding the importance of correct design in the front end arrangements of locomotives, current practice shows widely different ideas, and it is evident that much may be learned by such a thorough dicussion of the principles involved as is sure to be held at the meeting of the congress. It is learned from Mr. Quereau that 125 circulars have brought out but 20 replies, and when the advantages to be had from a thorough exposition of the reasons for the present diversity of opinion are appreciated the opportunity will be taken to assist the reporter to the utmost in representing American practice.

THE ATLANTIC TYPE AND THE WIDE FIREBOX.

By J. Snowden Bell.

It will no doubt be admitted, both by the designers and the builders of the two Atlantic type engines recently placed on the Chicago, Burlington & Quincy Railroad, and illustrated in the May issue of this Journal, that even if they shall, and it would seem probable that they will, fully and satisfactorily meet the conditions of exceptionally fast passenger service for which they have been constructed, it is within the possibilities that a still more effective performance might be attained by a modification of design which is not only fully applicable with the Atlantic type of engines, but it is also one with which that type would appear to have its most natural and logical application. The modification referred to is the employment of a wide, instead of a narrow, firebox, and the purpose of the writer is not in anywise to disparage the design as executed, but to invite attention to the special capabilities and advantages of this type of engine, in connection with a firebox of the former class.

The evolution of the "Atlantic" type has not been by any means a gradual one, it being a direct development of the essential features of two classes of engines which preceded it, neither of which has gone into use to any considerable extent. The first of these classes, which afterward came to be known as the "Columbia" type, originated, some time prior to 1875, in the practice of the Grant Locomotive Works of Paterson, N. J., and is illustrated in Plate XVIII. of the first edition of Forney's Catechism of the Locomotive. The engine there shown weighed 52,000 pounds, with 42,000 pounds on two pairs of 56-inch driving wheels, a pair of leading wheels and a pair of trailing wheels, each 28 inches in diameter, and 14x22-inch cylinders. This class was revived, with the enlarged dimensions and improved detail of the then existing practice, shortly before the Chicago Exposition, in which two examples, one, No. 694, P. & R. R. with a wide firebox, and the other, the "Columbia," with a narrow firebox, were shown. The first of these was styled by the builders, "Double Ender" type, and the second, "Special High Speed" type, although each of these designations would seem as fully and appropriately applicable to one engine as to the other.

The second antecedent class in the general line of the Atlantic type, comprised, it is believed, only three engines, each having a four-wheeled leading truck, a single pair of driving wheels, and a pair of trailing wheels. All three engines had wide fireboxes, and they were understood to have been designed with the special purpose of admitting of the use of a firebox of this class with driving wheels whose diameter was too great to allow such a firebox to be placed above them. The first of these engines, which was built in 1880, by the Baldwin Locomotive Works, was an admitted failure and went early into the scrap pile; the other two, which were much later in date, are in service with satisfactory results on comparatively light trains. These engines accord in general design with the Atlantic type, differing in detail only in having one pair of driving wheels instead of two, and the Grant engines and their successors differ in having a two-wheeled instead of a four-wheeled leading truck. The support of an overhanging firebox upon trailing wheels is a salient feature which is common to all three of the classes referred to, and it is obvious that it is entirely independent of the number of driving and leading truck wheels.

The first Atlantic type engine that was built has a narrow firebox and this feature of design has been frequently reproduced, among others in the C. B. & Q. engines, which are probably the latest. It may therefore be assumed to be presented by the builders as representative and good practice. There has been more or less discussion of the relative merits of the Atlantic and of the ordinary ten-wheel type, that is to say, that having six driving wheels and a four-wheeled leading truck, irrespective of the form of firebox applied in either case, but if the wide firebox be not employed, the object or reason of existence of the Atlantic type does not clearly appear, and it has not been made plain in any published statement within the knowledge of the writer. It will scarcely be contended, as a general principle, that the presence of trailing wheels is a desirable feature, particularly in an engine which is not intended to be run backward at speed, or that weight which is available for adhesion, (which, in the engines under consideration, is the substantial amount of 33,000 lbs.) should be transported as dead weight. Both these conditions are necessarily presented in the Atlantic type engine, and if a narrow firebox is used, are seemingly without any compensating advantage, either theoretical or practical. If, however, a wide firebox be employed, the Atlantic type is not only the ideal one for high speed service but is also apparently the only one in which practically any speed may be made and maintained which is per-

missible with present track, signal and brake facilities. With the application of the wide firebox, in connection with 84-inch, or larger driving wheels, the utility and necessity of the trailing wheels which characterize this type, are immediately and plainly apparent; the grate area and depth of firebox may be made as great as desired without interference with the driving wheels, and driving wheels of any desired diameter may be used without interference with the width and depth of the firebox. The sacrifice of the utilization of weight which is available for adhesion has its compensation, whether or not it be a sufficient one being, of course, a matter of opinion, in increased steam generating capacity and ability to employ lower grade and cheaper fuel. With the narrow firebox, the same sacrifice is made, in this type of engine, with no apparent compensation, and it is difficult to see why it should not be avoided by the substitution of a pair of driving wheels for the trailing wheels, which latter, though not desirable, may be tolerated when plainly advantageous, but should have no place in good practice unless their advantage is beyond reasonable doubt or question.

A number of Atlantic type engines have been constructed by the originators of the type, with wide fireboxes, and their performance in high speed service has been uniformly excellent and in some cases exceptionally so. Among these may be noted the Lehigh Valley engines running the Black Diamond Express, the C. R. R. of New Jersey on fast Royal Blue Line trains, and the Philadelphia & Reading engines on 50-minute trains from Camden to Atlantic City. It is, moreover, a matter of current report that wide firebox engines of this type are in preparation or under construction by the Pennsylvania R. R., and the adoption of this form of firebox by a system so conservative, and heretofore so pronounced in opposition to it, is strongly persuasive of its desirability in high speed service. All the wide firebox engines of this type which have been thus far built, as well as the majority of engines of other classes having this form of firebox, burn anthracite coal, for the successful use of which it has been recognized that a large grate area is practically indispensable, and we accordingly find 63.9 square feet of grate in the Lehigh Valley engines, and 76 square feet in the Jersey Central and Philadelphia & Reading.

It is unnecessary for present purposes, even if space permitted, to discuss the question of the relative merits and advantages of a large and a small grate, upon which much has been said and much remains to be said, more particularly in view of the results which may be reasonably expected in the performance of the new Pennsylvania engines referred to. It is, however, important to note that, doubtless by reason of the origin and general use of the wide firebox on roads employing anthracite coal as fuel, the impression seems to have been generally entertained that a wide firebox necessarily involves a material enlargement of grate area, and it is probable that this wholly unfounded assumption has contributed largely to the delay in the general introduction of the wide firebox into service with bituminous coal. It is obvious that a much smaller grate will be sufficient and proper with this class of fuel than with anthracite, and it is possible that, as has been contended by motive power officers whose opinions are entitled to respect, the area now employed in narrow fireboxes cannot be exceeded with economy or advantage. Be this as it may, the fact remains that a relatively long and narrow grate is, both in theory and in practice, inferior to one which approximates a square plan; the combustion in the latter can be effected more perfectly, and all parts can be reached by the fireman without the great and needless exertion which is involved in throwing coal to the forward portion of the long and narrow grates of present practice. While locomotive grates 10 to 12 feet long are fired every day and the engines taken over the road on time, it is manifest that the work required to do so is very much harder and more exhausting than would be the case if the same grate area was provided in shorter length and greater width. In the opinion of the writer, the special merit of the wide firebox lies not so much in the ability of obtaining an increased grate area, as in that of disposing any desired and determined area of grate in a more advantageous and desirable form, that is to say, approximately square.

The fireboxes of the C. B. & Q. engines are stated to be 10 feet long by 40⅜ inches wide, and the grate area as 33.6 square feet. The fireboxes of the Lehigh Valley engines are 9 feet 6⅝ inches long by 6 feet 8⅜ inches wide and the grate area is 63.9 square feet, and the fireboxes of the C. R. R. of N. J. and P. & R. engines are 9 feet 5⅝ inches long by 96 inches wide, and the grate area is 76 square feet. Assuming the grate area of the C. B. & Q. engines (in round numbers 34 square feet) to be the proper one under the conditions of fuel and service, it would seem that it could be provided with greater operative efficiency, in an engine of the same type, and otherwise identical in construction, by the employment of a firebox, say, 6 feet long by 5 feet 8 inches wide, the width proposed being a very moderate one, and the length such that it could be much more easily fired than that with which the engines are now running. Any desired increase of grate area within the present limit of length could, of course, be obtained by increase of width or length, or both, to which, as before indicated, the Atlantic type is perfectly and specially adapted. If, on the other hand, the embodiment of the present grate area in a long and narrow firebox is believed to be the better practice, the substitution of a pair of driving wheels for the present trailing wheels, and the consequent utilization of 33,000 pounds of weight for adhesion, naturally suggests itself. The objection may be raised to the wide firebox that its employment would involve an undesirable increase of tube length, the tubes being now 16 feet long. To this it may be replied, first, that the increase of length required would not be sufficiently great to be objectionable, and, second, that it could be avoided by the use of a firebox of the form which has been proposed by Mr. M. N. Forney, in which the portion next to the tube sheet is narrow enough to stand between the driving wheels and the remainder is widened beyond them as in the ordinary wide firebox construction. This form would present the further advantage of permitting the forward portion of the firebox to be used as a combustion chamber, a plan which has been practiced with satisfactory results in anthracite coal burning engines and which should operate equally well with bituminous coal. It is, moreover, exempt from the objections which are thought to exist to a combustion chamber located within the waist of the boiler, and which have limited its application to a comparatively small number of engines.

The addition of the C. B. & Q. Atlantic type engines to the number of new and interesting constructions which the increasing demands of high speed passenger service have recently brought into use, naturally suggests criticism and discussion of their special features of design. The views which are here expressed are rather in the nature of indicating that they might be better than they are, than that they are not as good as they ought to be, and except as to the view of the writer regarding the form of firebox, there does not appear, so far as can be ascertained from the published data, any particular in which their design and proportions seem susceptible of improvement. With their liberal heating surface of 2,510 square feet, ample cylinder power, perfected piston valve steam distribution and large driving wheels, it is reasonable to believe that the results of their performance will fully equal, and even exceed, the requirements for which they were designed, and these results will be awaited with interest by the railroad community.

The draught appliances have not been illustrated, but are presumably of approved type, and the handsome and symmetrical appearance of the engines will be admitted by every intelligent observer. The comparatively short front and taper stack are not inconsiderable factors in this regard, and form a pleasing contrast with the huge extended smoke-boxes of the C. M. & St. P. engines of the same type, the unsightliness of which is made more impressive by the growing conviction that they are correspondingly useless.

NOTES ON THE EFFECT OF PORTS AND VALVE GEARS ON THE PERFORMANCE OF HIGH SPEED LOCOMOTIVES.

By R. A. Smart.

In recent numbers of the "American Engineer" and other of the railway journals, a number of editorials and communcations have appeared on the general subject of high speed locomotives, in which emphasis has been laid upon the importance of having well balanced valves, heavy valve gears and large ports, in order that there may be no serious diminution of power when running at high speeds of rotation.

In order that a locomotive shall be able to maintain high speed with the modern passenger train, it should have such a perfected valve gear that the distribution of steam in the cylinder will be such as to develop the maximum power from every pound of steam used. The boiler has a very definite limit to its capacity and that limit is likely to be reached often with the majority of high speed locomotives. To secure the maximum effort, therefore, the valve gear should allow every pound of steam generated to count for as much as possible.

The difficulty usually encountered in securing a good distribution of steam at high speeds is two fold, (1) on account of insufficient port areas, which cause wire drawing, and (2) from badly balanced and heavy valves and light valve gears, causing serious distortion of the valve motion, both of which result in an uneconomical distribution of steam.

There is a tendency in some quarters to treat the subject of steam distribution as of small importance, the position being taken that so long as a locomotive pulls its train over the road, the question of steam economy is of comparatively small moment, and that if the cylinder power is insufficient, the remedy is to run with longer cut-off. That this position is not correct would seem to be the case when it is considered that given a locomotive running at high speed with a poor valve motion and consequently decreased power, and at the limit of its boiler capacity, the lengthening of the cut-off would not only increase the power of the locomotive as a whole, but might actually diminish it, whereas it might be possible to change the valve gear and ports so as to materially improve the steam distribution and thus increase its power by doing more work with the same amount of steam.

In order to show to what extent the locomotive valve motion may become distorted under ordinary conditions of running, two cards are shown in Figs. 1 and 2 which were taken from the same cylinder end under the same conditions, namely:

Speed ..... 25 miles per hour
Throttle ..... Fully open
Pressure in steam chest ..... 210 pounds

The full card (Fig. 1) was taken at the beginning of the run and shows the normal distribution of steam. The mean effective pressure of this card is 70.2 pounds. The smaller card (Fig. 2) was taken after about thirty minutes continuous running. Cards taken a short time previous to the one shown indicated the beginning of the distorted motion and the action represented in Fig. 2 continued until the run was stopped shortly after. Both cylinders were similarly affected and at the same time. The mean effective pressure of the smaller card is 59.4 pounds, a loss of 15.4 per cent. Similar action was noted at other times with the same engine under other cut-offs and for steam pressures above 200 pounds.

The engine from which the cards were taken has a flat, well-balanced valve, but somewhat larger and heavier in proportion than is often used. It is, however, fitted with an exceptionally strong and well made valve gear. No undue springing of the parts of the gear was observable while the small cards were being taken. The lubrication was steady and of greater quantity than would ordinarily be considered good practice. On taking off the steam chest cover immediately after shutting down, the parts were found to be in good condition and well lubricated. The lubricant was evenly distributed and there was no evidence that the oil had collected in the pipe and dropped down when the throttle was closed.

The distorted valve motion was without doubt largely due to the use of a flat valve with high steam pressure. While a heavier valve gear and more lubricant would probably have reduced the distortion materially, no blame should be atttached to those features, since they were designed to amply meet the conditions of ordinary service. This case is perhaps an exceptional one, but it shows what may happen under conditions which would ordinarily be considered favorable to good results, and serves to call attention to the importance of having light and properly balanced valves and strong, well made valve gears in order to secure a proper steam distribution.

Mention has been made of the importance of using large ports in order to avoid excessive wire drawing. The following tabular statement shows what may be accomplished by this means in the way of increasing the mean effective pressure while retaining the same percentage of cut-off, release and compression.

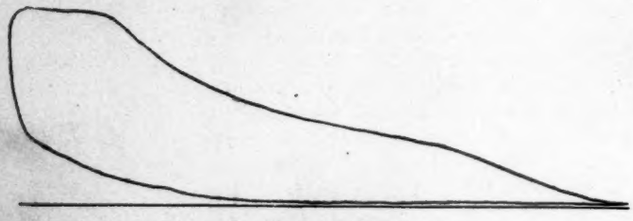


Fig. 1.—Left Head End.
Mean Effectivv Pressure, 70.16. Scale, 150.

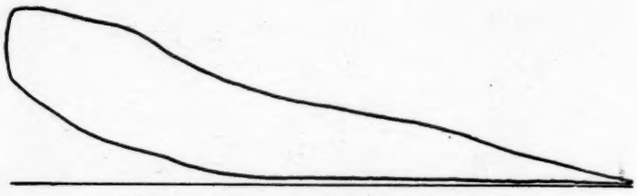


Fig. 2.—Left Head End.
Mean Effective Pressure, 59.44. Scale, 150.

	Cut-off.	R. P. M.	Initial Pres.	M. E. P.	P. C. Increase
Engine No. 1...	43.8	184	116	49.7	....
Engine No. 2...	42.3	194	111	59.3	19.3
Engine No. 1...	25.1	187	115	30.8	....
Engine No. 2...	25.8	193	119	47.4	53.8

The cards taken from the two engines under the conditions shown in the table were similar in every respect except as to the relative pressures. The increase in mean effective pressure given by engine No. 2 was wholly due to the use of an Allen valve and larger ports in proportion to the cylinder volume.

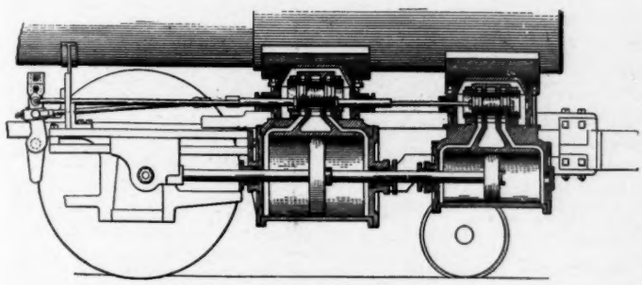
What has gone before serves to show the importance of careful attention to the design and construction of valve gears and ports, and to emphasize the relative importance of these details in their effect on capacity of the locomotive at high speeds. There is, however, one important consideration which should not be overlooked when an attempt is made to secure large ports. An increase in port area means generally an increase in the percentage of clearance which, it is usually conceded, should be kept as low as possible. An increase of but a few per cent. in the clearance above that which is ordinarily considered to represent good practice may result in a very marked decrease in the economy of the engine, thus losing all of the benefit which would be expected from the increased size of the ports.

PLAYER'S TANDEM COMPOUND LOCOMOTIVE.

The accompanying diagram illustrates the chief features of a new arrangement of compound cylinders recently patented by Mr. John Player, Superintendent of Motive Power of the Atchison, Topeka & Santa Fe Railway. Several engines have been built from drawings embodying this construction, and we are informed that the results on the road are satisfactory.

The cylinders are in two sets combined with half saddles and attached to a long smoke-box ring with sufficient space between the cylinders to admit of taking out the rear or low pressure piston without removing the high pressure cylinder, the piston rods being spliced for this purpose. Piston valves are used and an interesting feature of their construction is the absence of packing rings. The valves as first arranged had packing rings of the usual form and these were afterward replaced with narrower ones, until finally, on account of wear and friction, they were discarded altogether and the valves were made of cast iron in the form of plugs absolutely without packing.

The drawing does not accurately represent the construction of the locomotives, but it illustrates the principles. The real novelty in the plan is in the arrangement of the valve rods whereby the cut-off on the high pressure cylinder may be changed by a simple adjustment without interfering with that of the low pressure cylinder. The valve stem for the low pressure cylinder is hollow, and that of the high pressure cylinder passes through it. The valve rods have separate attachments to the upper rocker arm and by changing the



Player's Tandem Compound.

point of connection of that for the high pressure valve its travel is altered and the cut-off may be made suitable for the special conditions under which the engine is to work. It is clear that this arrangement of tandem cylinders avoids the extremely large steam clearances of some former designs of tandem compounds and this valve gear is not at all complicated. It is an ingenious and interesting plan, and if it demonstrates the possibility of using piston valves of the plug type and shows that the cut-off of the high pressure cylinder may be made adjustable as in marine practice it will attract a great deal of attention. It seems likely to accomplish these results.

Fearless discussion of suggested departures from current practice, such as is held before the American Railway Master Mechanics' Association, is thought by our English contemporary, "The Mechanical World," to explain American leadership in locomotive matters.

The claim made by Mr. Tripler that he makes ten gallons of liquid air by the use of three gallons has been answered by President Henry Morton by showing that Tripler produces a gallon of liquid air by the use of 10 horse-power hours, whereas, even in an ideally perfect engine, only three-quarters of one horse-power hour may be obtained from this quantity of liquid air when used as a source of power.

THE CONSTRUCTION OF A MODERN LOCOMOTIVE.

By T. R. Browne, Master Mechanic Juniata Shops, P. R. R.

The Arrangement and Division of Labor.

VIII.

Probably no other element in the construction of the modern locomotive or in general manufacturing has a more direct bearing upon the fluctuations of cost as connected with the management of a plant than the character, division and management of the labor employed. Even the best and most modern machinery and general equipment will utterly fail of its purpose with indifferent and unskillful handling and we consider the labor, in which we include the men actually employed on production, therefore, a very special part of the equipment of the plant, and their selection and management, considered from the standpoint of equipment, must receive the same general order of consideration as that used in selecting the appliances which they are to operate.

It is obviously unprofitable to employ a high grade of mechanical labor in the operation of machines when those machines may be equally well operated by a lower grade of labor and at much less cost. The great improvements that are being made in facilities for general manufacturing render the employment of a high grade of labor for their operation unnecessary, although many times, we think, this point is overlooked, and a skilled mechanic is employed to operate a machine when a much lower grade of labor would answer equally well. It is upon the amount of labor charged against a given amount of output that the profits or earnings to cover expenses of operation and other expenses of the plant are charged, and the character of this labor must be such as to produce the largest amount of output at the minimum cost.

Taken generally, the plant may be divided into two parts: The producing portion and the non-producing portion. As a result of the operation of the first, the profits and income must be such as to pay the expenses of the second. In the producing portion of the plant each machine must, by reason of its earnings, contribute a proportion of the expense of its operation, repairs, etc., as well as a proportionate amount of the cost of maintaining the balance of the plant and its operating expenses. It will be obvious that, in proportion as the original cost of a machine is greater, the proportion of the expenses of the plant which it shall bear will be increased and the necessity for its economical and profitable operation become more emphatic. The value of the output from this machine, independent of expenses, is represented by the value of of the raw material, plus the labor of completing it; and this labor alone, in most cases, must earn the proportion of the expenses of operating and maintaining the plant which are charged against this machine.

The general arrangement and ranking division of all labor throughout the plant should be such as to encourage a regular system of promotion from the lowest grade to the highest, and in accordance with the ability of the man. It has been found that the larger part of the labor employed can be considered as specialists, or a class of men who do not have a trade, but have risen from the ranks of lowest labor by reason of peculiar fitness for some special line in which they have become more or less perfect; and the policy of selecting from the most ordinary laborers employed men who are trained as specialists has been found most successful in providing not only efficient and careful employees on the line of work which they are specially trained for, but a force which is contented and reliable. In their rank as specialists they start at a rate of pay which is slightly more than that paid the ordinary laborer, and from time to time are increased in their rating as they become more proficient; length of service also has something to do with the question of rate. They do not, however, ever reach the regular rate paid the skilled mechanic. In referring to skilled labor, that class of labor or mechanics are meant who are capable of performing any kind of work in their line of trade.

There is a class just below this who are not quite as high up in their trade and who can very properly be ranked as the highest order of specialists, and for the sake of illustration, we have assumed this latter class as included in the so-called specialist. Another source of supply for renewal of this class of labor is from the ranks of young men who are too old to learn a trade, and yet below age and not entitled to employment as full-grown men. This class of young men are placed on machines or various work throughout the plant, and a careful record kept of their performance and skill, and at least every six months tried on a new class of work, not only for their encouragement, but for the purpose of determining their best efficiency. These young men eventually become specialists in similar manner to the class of men above referred to, and this system frequently develops what proves to be a very valuable man.

Where piece-work is generally adopted, both of these classes of labor work on a piece-work basis, in which case their earnings are considerably more than their regular rate by the day. It is pretty well recognized that in neither a railroad or locomotive shop, where the construction of locomotives constitutes the main industry, can any one of the trades be thoroughly learned, at least in that sense that a man who has learned his trade under those circumstances is thoroughly fitted to compete with the highest grade of mechanic, who can take up practically any kind of work, or who has learned his trade in a shop where the character of the work is much finer and closer than that required in the construction of the locomotive. It is, however, true that the experience which the apprentice will gain in a locomotive shop is a most valuable introduction to the trade which he has adopted and to further experience in the same trade at other shops.

In connection with the construction of the locomotive there are practically eight trades represented, this division of trades representing approximately an equal number of departments in which the work in this line is carried on. The advantage of having a trained force in each one of these departments will be appreciated, and the arrangement of apprentices, from which the supply of skilled labor to a very large extent is drawn, is such as to give them a training in such of these trades as they may select, and while the complete trade, as above intimated, cannot be taught them, at least so much of it as may be useful at the plant and make them valuable as employees may be acquired; and it is from the best of these apprentices in each one of these trades that the skilled labor is generally drawn to fill that particular rank. The men employed throughout the plant should be assured that from the lowest position they can be promoted and rise to the highest directly in proportion to their skill and faithfulness, and, independent of the fact that they do not possess a regular trade, can occupy a position, by reason of their skill in one direction, and which will pay them almost, if not quite, the rate of the mechanic. This has the effect of continued encouragement to advance themselves by every effort in their power.

Independent of the operation of the various facilities throughout the plant, or that type of work on which the specialist can be employed to best advantage, there is the repair work, special work, and even work in connection with actual production, which requires the services of the most skilled mechanics, the object being to use this class of labor for maintaining the equipment which specialists are used to operate and such general equipment throughout the plant as may be incidental to its operation. In a general sense, the division of labor throughout the plant consists of the ordinary or lower grade of labor used in connection with the handling of material and general work requiring such labor; the specialists or shop hands, who are employed directly in production; the apprentices, from whom are drawn the supplies of skilled labor or mechanics, and the skilled mechanics, who are used on that character of work requiring their peculiar skill, and which cannot be performed with the same economy by specialists.

Grouping the ordinary laborer, specialist and the apprentices together, they constitute fully 82 per cent. of the labor employed, the remaining 18 per cent. being represented by the skilled mechanics. It is obvious that the percentage of skilled labor required for this class of work will vary somewhat with the nature of the work, and is therefore different in different departments. Considering only the four principal shops, owing to the fact that the largest amount of labor is employed in these departments, the proportion of specialists or unskilled labor as compared to skilled labor in each one of these departments, these proportions being intended to represent numbers of men and not amounts of money, would be about as follows: In the blacksmith shop it will amount to 70 per cent., as compared with 30 per cent. of skilled labor; in the boiler shop, 80 per cent., as compared with 20 per cent.; in the machine shop, 88 per cent. and 12 per cent.; and in the erecting shop, 91 per cent. and 9 per cent.

The writer is convinced that the question of distribution of labor and its character, in proportion to the character of work to be accomplished and the demands of each particular case, will amply repay the closest study, and that the amount of economy to be derived by a proper arrangement of labor will entirely offset the additional amount of salary which the character of skill and ability on the part of the foremen to look after a matter of this kind would command.

The division of labor by operations to be performed must, to a very large extent, be dependent on the methods to be followed in construction, and will also be controlled by the character of facilities and the arrangement of shop. The ideal condition would, of course, be one in which a division of the whole locomotive could be so made as to bring along every one of its parts at or about the same time and without interefering with any part of the general work.

The arrangement of the division of the force by operation will also to a very large extent depend upon the personal ability of the foreman in charge of the department; and the best man in this respect, at possibly the highest rate of pay, is not too good, in our estimation, to accomplish the results which can be brought about by a careful consideration of this whole subject.

PERSONALS.

Mr. James Thompson has been appointed Superintendent of Motive Power of the Pennsylvania Division of the New York Central to succeed Mr. W. A. Foster.

Mr. C. F. Winn has been appointed Master Mechanic of the Chesapeake Beach Railway, with headquarters at Washington, D. C.

Mr. G. R. Brown has retired from the position of General Superintendent of the Fall Brook Railway, owing to the absorption of the road by the New York Central.

Mr. A. C. Deverall, Superintendent of the St. Cloud shops of the Great Northern, has been promoted to the position of Superintendent of the car and machine shops at St. Paul, to succeed Mr. S. F. Forbes, promoted.

Mr. G. L. Warren, for the past fifteen years Chief Draftsman for the Chicago, St. Paul, Minneapolis & Omaha, has been appointed Mechanical Engineer.

The Baltimore & Ohio has appointed Mr. W. C. Hayes locomotive superintendent. He will have charge of all enginemen and engines while on the road and will instruct the men in the handling of the engines, but will refer questions of discipline to the Master Mechanics. He will have charge of the traveling engineers. The traveling engineers will examine and employ firemen under the customary physical and educational requirements. This plan places the operation of the locomotives in the hands of one man and much may be expected from this in the way of uniformity of instruction and service.

Mr. Schuyler Hazard has resigned as Principal Assistant Engineer of the Cleveland, Cincinnati, Chicago & St. Louis Railway, to engage in partnership with Mr. Thomas Lee of Cincinnati in the sale of Lumen bronze and anti-friction metals, as noted elsewhere. These bearing metals are the invention of Prof. R. C. Carpenter of Cornell University. Mr. Hazard is a member of the American Society of Civil Engineers and is a young man to have held the important position in the "Big Four" that he now leaves. He graduated in civil engineering from the Massachusetts Institute of Technology in 1890, and almost immediately entered the service of the P. C. C. & St. L. Railway at Pittsburgh. He soon after went to the "Big Four" as first assistant engineer of the Peoria division. His next advance was to take the position of assistant engineer of the Cincinnati and Cleveland divisions. Then he was made engineer of maintenance of way of the Cleveland division, after which he was put in charge of construction. His appointment to the position he has just resigned followed quickly after the others mentioned. Mr. Hazard has a wide acquaintance and the ability he has shown in railroad work promises success in his new undertaking.

Charles M. Higginson, assistant to the President of the Atchison, Topeka & Sante Fe Railway, died at his home in Riverside, Ill., May 6, after a brief illness. He was a valuable officer and contributed in a very important way to the success of the present administration of the Atchison properties. Another assistant to the President may be appointed, but it will be extremely difficult to fill his position because he was fitted by rare perception, searching insight, patient perseverance, wide experience and unusual foresight in the administration of the details of railroad operation. His was a modest, retiring disposition and he was sometimes misunderstood, but those who knew his work and his chief idea in the consideration of a business question—"What is there in it for the Atchison road?"—were bound to admire him and his devotion. His work was so well outlined and so thoroughly organized all over the system that the properties will continue to reap the benefits, and this is a good test of its character and quality. Mr. Higginson began railroad work in 1865 in the engineer's department of the Burlington & Missouri River R. R. He was appointed Purchasing Agent of the Toledo, Peoria & Western in 1875, the following year he became Purchasing Agent of the Chicago, Burlington & Quincy, and in 1879 was made Assistant Auditor with special charge of statistics. In 1890 he became Assistant to the Vice-President, and Mr. E. P. Ripley appointed him as his assistant upon taking the presidency of the Atchison. Mr. Higginson was born in Chicago in 1846 and received his education at the Lawrence Scientific School of Harvard University. He was interested in scientific studies and was a recognized geologist. He was a member and for several years President of the Chicago Academy of Sciences, he was one of the Board of Managers of the Young Men's Christian Association in Chicago and a Fellow of the Royal Society of Great Britain. He will be missed by many and not least by the young men whom he had selected for positions in the various departments of the road.

Captain Alfred E. Hunt, one of the founders of the Pittsburgh Testing Laboratory and President of the Pittsburgh Reduction Company, died suddenly at the Hotel Lafayette, Philadelphia, April 26, from illness contracted during his service in Porto Rico in the recent war. Captain Hunt was very prominent in metallurgical matters. He was born at East Douglass, Mass., in 1855, and was educated in the schools of Boston, graduating in mining and metallurgical engineering from the Massachusetts Institute of Technology in 1876. His first professional work was with the United States Geographical Survey, after which he was connected with the Bay State Iron Works at South Boston, where the second open hearth steel furnace in this country was erected. From there he was sent to Michigan to explore for iron ore, and his report led to the opening of the Michigamme mines and the development of the Wisconsin and Michigan iron regions. Next he went to Nashua, N. H., as manager and chemist of the open hearth steel works there and later became chemist and superintendent for Park Brothers, Ltd. In 1882, with Mr. George H. Clapp, he formed the firm of Hunt & Clapp for the purpose of testing and reporting upon materials and the inspection of iron and steel structures. The business grew rapidly and after ten years a stock company was formed, which is now so widely known as the Pittsburgh Testing Laboratory, Limited, and is recognized as the leading establishment of its kind in the country. In 1888 Captain Hunt began the production of aluminum and organized the Pittsburgh Reduction Company to manufacture it on a large scale. The wide introduction of this metal, which was due to the reduction of the cost of production, is attributed chiefly to him. His standing as an engineering, chemical and metallurgical expert was high and the courts frequently required his testimony in difficult cases. He wrote a number of valuable scientific papers, which have been published in the proceedings of the national technical societies. He was a member of the American Society of Mechanical Engineers, the American Society of Civil Engineers, a fellow of the American Society for the Advancement of Science, a member of the Iron and Steel Institute of Great Britain and of other prominent technical organizations. He was prominent in military matters and was Captain of Battery B of the National Guard of Pennsylvania, and during the recent war he led his command on the expedition to Porto Rico. Captain Hunt was successful in his business undertakings and he will be missed by business and professional associates and by a very large number of friends.

EQUIPMENT AND MANUFACTURING NOTES.

The Ajax Metal Co., of Philadelphia, have not entered the combination of firms of manufacturers of bearing metals, and do not propose to do so. This concern will continue to produce superior bearing metals, as in the past, at the lowest possible prices and entirely independent of the trust.

J. A. Fay & Egan Co., the largest manufacturers of woodworking machinery, whose plant is located at Cincinnati, Ohio, have demonstrated that prosperity has returned in their line of business by building a large addition to their plant, which is now being equipped with the most modern tools. May 1st the wages of employees were raised 10 per cent. without solicitation from the men, which is the best possible indication of a satisfactory condition of business and a return of "good times."

Mr. Daniel M. Brady, for the last eleven years President of the Brady Metal Company, has severed his connection with that firm and has organized the Brady Brass Company, with general office and works at 202 Tenth St., Jersey City. The new company manufactures high grade castings and metals for bearings and other details of steam and street railroad work. Mr. A. Onslow, formerly for seventeen years Superintendent of the D. A. Hopkins Self Fitting Lead Lined Journal Bearing Company, and for eleven years Superintendent of the Brady Metal Company, has associated himself with the Brady Brass Company as General Superintendent.

The Graham Equipment Company, of Boston, have received an order for 135 auto-trucks from the National Transportation Company, of that city, for use in the construction of equipment for various bus lines which are being established in the suburbs. They will have steel frames and will be equipped with Graham's spring suspension and equalized brakes and rubber tires will not be used. Another order has been received for 12 combination smoker and baggage cars, five 10 ton steel coal cars and five flat cars of the same capacity for an electric road in Massachusetts. They will have Graham's steel underframes, made of merchant shapes, secured by bolts, and his spring suspension and equalized brakes. This company is building two sample trucks for the Goodwin Car Co. for exhibition at the June conventions.

The Magnolia Metal Company have opened an office in Montreal, Room 524 Board of Trade Building, for the convenience of their Canadian business.

C. L. Berger & Sons, successors to Buff & Berger, instrument makers of Boston, have just purchased the famous Wardeman automatic circle-dividing engine, which graduates a circle to 5 seconds of arc within a limit of error of one second. There are only two in this country to-day which can do such fine work. On account of their increasing business they have been obliged to enlarge their works, increase their force, and add materially to their machinery. April 29 they celebrated by appropriate ceremonies the completion of their three thousandth instrument. Their catalogue, just issued, is a hand-book and treatise on the construction and care of instruments of precision, full of valuable hints and information.

In the earlier days of the use of Russia sheet iron in this country its cost per pound ranged as high as 25 cents. In fact, up to the time that Wood's patent planished sheet iron was invented and put upon the American market, and for a short time afterward, the market price varied from 15 to 25 cents. Not long after its advent the American product was demonstrated to be superior to Russia iron in quality, uniformity of color, gauge, and finish. Moreover, it was placed upon the market in dimensions to suit requirements. And not only did it possess superiority of quality, but its price was fixed at a figure much lower than that of Russia iron. Wood's patent planished sheet iron has driven Russia iron out of the market, and it costs the consumer rather less than one-half the price formerly charged for Russia iron.

Mr. Thomas Lee, of Cincinnati, and Mr. Schuyler Hazard, formerly Principal Assistant Engineer of the Cleveland, Cincinnati, Chicago & St. Louis Ry., have formed a partnership, with offices at 209 Race Street, Cincinnati, for the sale of the Lumen metals in the States of the middle west, the south and the southwest. "Lumen" was discovered and perfected by Prof. R. C. Carpenter, of Cornell University, and upon subjecting it to thorough laboratory tests it was put on the market and has been adopted and is used by a number of prominent engineering firms. It melts at 680 deg., has a specific gravity of 6.9; tensile strength, 31,000 lbs.; compressive strength, 70,000 lbs.; its coefficient of friction is 0.00625, and it may be poured like babbit.

The Chicago Pneumatic Tool Co. has purchased the patents formerly owned by the Consolidated Pneumatic Tool Co., now defunct. These patents include all the Keller and Wolstencroft types of tool construction and in addition several new applications which have not yet been taken out. These patents originally cost the Consolidated Pneumatic Tool Co. about $40,000. Mr. Sidney H. Wheelhouse, formerly with the M. M. Buck Manufacturing Co., St. Louis, has become connected with the Chicago Pneumatic Tool Co. as southwestern agent, with office and salesroom at 409 North Fourth Street, St. Louis. The satisfactory condition of the business of this firm is shown by the large number of orders received. We are informed that May 8, at the Chicago office, the mail orders called for 81 pneumatic hammers and riveters, 57 drills of different sizes, and 17 miscellaneous tools, a total of 155 tools ordered in a single day. Among recent orders is one for eight penumatic riveters, for Shanghai, China. The Chicago Pneumatic Tool Company and Joseph Boyer have begun suit in the United States Court at Philadelphia against the National Pneumatic Tool Company for infringement of patents covering the Boyer pneumatic hammer.

The Pressed-Steel Car Company has arranged to have a comprehensive exhibit of its various styles of cars at the Old Point Comfort Conventions. Not only will their types be shown in new cars, but a complete series of cars manufactured early in the history of the company and since then in use by prominent roads, will be exhibited. The cars will be sent to Old Point Comfort in exactly the condition in which they are received from the roads, in order that there may be, for the benefit of railroad men, an ocular demonstration as to how the cars wear in service. Of the old cars to be on view the Pennsylvania Company will contribute one; the Baltimore & Ohio, one; the Pittsburgh, Bessemer & Lake Erie Railroad, one self-clearing hopper, and the Lake Superior & Ishpeming Railroad, an ore car. The exhibit will be not only interesting, but instructive.

The firm of Manning, Maxwell & Moore, and its kindred interests, the Ashcroft Manufacturing Co., The Shaw Electric Crane Co., The Consolidated Safety Valve Co., The Pedrick & Ayer Co., and the Hayden & Derby Manufacturing Co., whose New York offices and salesrooms have been located a great many years at Nos. 111 and 113 Liberty street, have removed to the new Singer Building, 85, 87 and 89 Liberty street, corner of Broadway. The change has been found necessary owing to the steady increase of business done by the firm. The new quarters are large and convenient, and arranged to meet the situation fully. The ground floor will be devoted to show rooms and has facilities for carrying a complete stock of everything pertaining to the business. The first floor will be occupied entirely by the business offices, which are complete in every detail, being fitted throughout with the most modern appliances to enable the large volume of business transacted to be handled conveniently, economically and expeditiously.

"Axle Light" equipment, furnished by the National Electric Car Lighting Co., 71 Broadway, New York, has been ordered by President Truesdale of the Delaware, Lackawanna & Western, for 15 new passenger coaches now building at Dayton, Ohio. They are to run in suburban service. The Atchison, Topeka & Santa Fe have also contracted for 34 cars in addition to the 88 already equipped with electric lighting, the new cars to run between Galveston and Houston, Texas. Among other recent installations are the Illinois Central officers' Car No. 2, now on a trip to Mexico, and the private car "Emmalita," of Mr. George Crocker, Vice-President of the Southern Pacific Company. This car recently carried Mr. Crocker and a party of friends from New York to San Francisco. It was in the charge of the colored porter, who gave all necessary attention to the lighting equipment and, we are informed, abundance of light was furnished with power to spare for running electric fans.

The Third New York Electrical Exhibition opened at the Madison Square Garden May 8 with the usual simple ceremonies, including an address by Senator Chauncey M. Depew. The exhibition will be open until June 3, and during its progress the New York Electrical Society will hold several technical meetings. The exhibition is held under the auspices of the National Electric Light Association and the annual convention will be held during the exhibition. The chief feature of the exhibition is to show progress in automobile carriages, of which a large number of admirable examples are included. A collection of electric lamps arranged chronologically is exceedingly interesting, and among the novelties are wireless telegraphy and the use of a searchlight to transmit speech. The exhibit of direct connected motors for machine tool driving and that of the signal corps of the United States Army are specially interesting. There are fewer exhibits than last year and the selection has been made with care.

The Pneumatic Supply & Equipment Co. has been organized under the laws of the State of New York and has opened an office at 120 Liberty street, New York. It is the purpose of this company, as its name implies, to deal generally in compressed air equipment and it will make a specialty of the installation of complete plants, eliminating the division of responsibility which has heretofore existed. The company is bringing out several specialties in the compressed air line, such as pneumatic oil rivet forges, quick acting hose couplings, and has in addition arranged agencies for several standard types of compressors. Mr. J. W. Duntley, the President of the Chicago Pneumatic Tool Co., is the President of the new company. Mr. E. B. Gallaher, formerly with Messrs. Patterson, Gottfried & Hunter, is Vice-President and Engineer, and Mr. W. P. Pressinger, formerly Manager of the Clayton Air Compressor Works, is Secretary and Treasurer. The rapidly widening scope of compressed air application opens a large field of usefulness for the new company, and the character of its incorporators is an assurance of its ability to meet the requirements.